DESIGN OF AN ELECTROHYDRAULIC WAVE FOLLOWER SYSTEM

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THESIS

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WAVE FOLLOWER SYSTEM

by

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System modeling showed that the wave follower provided more than sufficient cylinder amplitude response for the design payload in the frequency design range. The phase response became degraded at higher frequencies. However, this could be corrected by incorporating a lead/lag section in the controls portion of the system.

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Design of an Electrohydraulic

Wave Follower System

by

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Lieutenant, United States Navy
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NOMENCLATURE

A	area [IN^2]
A_p	piston annular area [IN^2]
$\bar{A}_p$	piston acceleration [IN/SEC^2]
c_d	orifice coefficient
$c_{tp\ell}$	total cross port leakage [IN^3/SEC]
d	diameter
e_{CPI}	cylinder position indicator signal [V]
e_E	error voltage [V]
e_S	error signal [V]
e_{WG}	wave gauge signal [V]
F	frequency [HZ]
g	gravitational acceleration [IN/SEC^2]
h_L	head loss coefficient
h_p	pressure loss coefficient
I	moment of inertia [IN^4]
K	combined gain
K_a	amplifier gain
K_c	valve pressure coefficient [$IN^3/SEC/PSI$]
K_{ce}	total valve pressure coefficient [$IN^3/SEC/PSI$]
K_{co}	null valve pressure coefficient [$IN^3/SEC/PSI$]
K_p	valve pressure sensitivity [PSI/IN]
K_q	valve flow gain coefficient [$IN^3/SEC/IN$]
K_{q0}	null valve flow gain coefficient [$IN^3/SEC/IN$]
K_W	wave gauge gain [V/IN]

ℓ	length [IN]
L	wavelength [IN]
m	mass (LBM]
m_T	total mass [LBM]
P	pressure [PSI]
P_L	load pressure [PSI]
P_R	return line pressure [PSI]
Q	flow [GPM]
S	LaPlace variable
S.F.	safety factor
T	thrust load [LBF]
V	velocity [IN/SEC]
W	valve gradient [IN^2/IN]
W_O	oil weight [LBF]
W_P	payload weight [LBF]
W_R	rod weight [LBF]
W_T	total weight [LBF]
X_P	piston pressure [IN]
X_W	wave position [IN]
β_ℓ	equivalent bulk modulus [PSI]
δ	wave steepness
δ_h	hydraulic damping
τ_c	time constant [SEC]
ρ	density [LBM/IN^3]
ω	radian frequency [RAD/SEC]
ω_h	hydraulic natural frequency [RAD/SEC]

I. INTRODUCTION

A. REASONS FOR A WAVE FOLLOWER

All energy transferred from the atmosphere to the ocean is transmitted across the air sea interface. The actual phenomena of these transport processes are not well understood. An active wave follower system capable of positioning instruments at a constant relative height above and below this interface would be valuable for investigating the small scale energy exchanges occurring at this interface.

The large scale effects of this energy transfer are of utmost importance. Wave height and frequencies, ocean currents, ocean heating and resultant thermal effects upon acoustic propagations are some of the direct consequences of this air sea interaction. Additionally, data describing the near surface refractive index could be gathered with an active wave follower. These data are of particular importance in determining the behavior of both electro-optic and electro-magnetic transmissions in this near-surface region.

Data can be gathered in the laboratory concerning certain of these effects. However, great difficulty is found in overcoming the problems of extrapolating these data to match actual open ocean conditions. These difficulties arise from scaling problems and wave tank induced pressure distortions. The use of an active wave follower capable of carrying and accurately positioning sensors at the air sea interface in the ocean is therefore most desirable.

B. PREVIOUS DESIGNS

Wave followers of various types have been developed periodically over the past ten years. Most of these can be classed into two design modes. The first and most basic uses a float with instruments attached directly to the float. The second design involves an electromechanical or electrohydraulic positioning system. These positioning systems are either fixed to a stationary tower or are anchored to the ocean floor. A more comprehensive background on existing wave follower designs is found in Bonnett [2].

C. PRESENT DESIGN

The design of the present wave follower is a synthesis of the two basic types. The electrohydraulic (active) section of the wave follower is fixed to a float (passive) section. The passive section responds to low frequency relatively high amplitude waves, and the active section responds to the higher frequency (up to 10.0 HZ) lower amplitude waves. This design overcomes the inherent drawback of the simple float type--inability to respond to high frequency waves--and the limitation on stroke for those systems attached to fixed frames.

II. SYSTEM DESIGN PARAMETERS

A. DESIGN PAYLOAD

The weight of the system payload is a direct function of the instrumentation packages affixed to the power cylinder. The type of instruments loaded are determined by that data which is desired. Typical instrumentation arrays for the system consist of hot film/wire anemometers, wave gages, thermistors, pressure sensors, salinity probes and humidity sensors. The weights of these individual sensors range from a few ounces to about two pounds. However, each of these instruments carries with it not only the sensor but also the power and data cables required in its operation. Therefore, a fairly substantial increase over sensor weight must be supported by the unit.

The system was designed for a static load weight of 25.0 LB. Calculations to determine required pressure used a load safety factor of two. In essence, the system is capable then of supporting a load of 50.0 LB.

Since the system was to move these weights in response to ocean motions, there had to be sufficient power available to accelerate the static load. The maximum acceleration used in the design was 32.2 FT/SEC^2 (1g). Accelerations of greater magnitudes are not expected in dealing with the deep ocean gravity waves for which this system was designed.

B. FREQUENCY AND AMPLITUDE SPECIFICATIONS

The basic wave types to which the system was designed to respond are the deep ocean gravity and capillary waves which have frequencies up to 10.0 HZ. The theoretical maximum steepness of $\delta = 1/7$ where $\delta = H/L$

(See Figure 1) was used to relate wave length to wave height. These results were then manipulated to yield wave follower piston travel and flow demand. The wave relationships used are shown below, and the calculated results are listed in Table I.

Deep water wave:

$$L = \frac{g(1/F)^2}{2\pi}$$

$$\delta_{max} = \frac{1}{7}, \quad H = \frac{2}{7}$$

The last column of Table I is entitled flow (Q). The flow requirement was determined using a standard 1.0 IN diameter cylinder with a 5/8 IN rod yielding an annular area of 0.4786 IN^2.

It can be seen from Table I data that a marked increase in flow demand occurs at 1.0 HZ. This increasing flow demand reflects itself in a requirement for a heavier hydraulic pump and electric motor. The increased weight of these components would have had a negative effect on the design. Therefore, some compromises in system response became necessary.

TABLE I: Wave Profiles and Required Flows

Frequency F (HZ) (SEC^{-1})	Wave Length L (IN)	Wave Height H_{max} (IN)	Flow Q (GPM)
1.0	61.50	8.79	2.18
1.5	27.33	3.90	1.45
2.0	15.37	2.20	1.09
2.5	9.84	1.41	.87
3.0	6.83	.98	.73
3.5	5.02	.72	.63
4.0	3.84	.55	.55
4.5	3.04	.43	.48
5.0	2.46	.35	.44
5.5	2.03	.29	.40
6.0	1.71	.24	.36
6.5	1.46	.21	.34
7.0	1.26	.18	.31
7.5	1.09	.16	.30
8.0	.96	.14	.28
8.5	.85	.12	.25
9.0	.76	.11	.25
9.5	.68	.10	.24
10.0	.61	.09	.22

III. DESIGN PHILOSOPHY

The first decision made concerned the basic configuration of the system. There were two competing designs, electromechanical and fluid power (hydraulic or pneumatic). The considerations listed below were made to qualify the inherent advantages and disadvantages of fluid versus mechanical systems and hydraulic versus pneumatic systems.

Advantages of Fluid Power Systems Compared to Electromechanical Systems are:

1. better power to weight ratio
2. better rigidity in steady state (liquid)
3. better speed and directional response
4. more compact construction
5. minimum of backlash
6. accurate control of velocity and position easier to accomplish
7. lower wear rate in operation

Disadvantages of Fluid Power Compared to Electromechanical Systems are:

1. more susceptibility to contamination (moisture and particulate)
2. requires fluid storage (accumulator, tank) components
3. more costly construction

Advantages of Hydraulic Systems Compared to Pneumatic Systems:

1. stiffer system response
2. better response speed at same power level
3. less inherent friction-due to lubricating properties of the liquid
4. less susceptibility to moisture and freezing problems
5. easier maintenance of seals
6. less quiescent power drain
7. usually better efficiency
8. better position and velocity control; no gas compressibility effects

Disadvantages of Hydraulics Compared to Pneumatic Systems are:

1. special fluid and reservoir requirements
2. viscosity temperature dependence of working fluid more severe

The hydraulic design was chosen as preferable for two basic reasons: First, more accurate control of position (eg., no backlash) was possible; and second, greater power to weight ratio (minimum size for the same instrument payload) was feasible.

Since environmental parameters were to be measured by instruments affixed to the system, a minimum of system self-generated disturbance was desired to preclude unwanted local changes in the measured quantities. The system was to be added to an existing float and used in conjunction with existing data gathering instruments on this common float. It was therefore necessary to minimize adverse effects on the original configuration. Both of these requirements could be best achieved by minimizing the size and frontal area of the wave follower system.

These overriding concerns, along with a desire to meet system specifications of amplitude and frequency response, resulted in the final system design. The system container is smaller than the outward dimensions of the existing float, thereby actually reducing the above water disturbance. In the final configuration, the cylinder with its instrumentation package will be placed two and a half feet away from existing sensors and will be located off to the side (90°) from the forward direction. This arrangement is shown as Figure 2.

Additional considerations included concerns about the use of the system in a marine environment. Thus questions about water tight integrity, hydrostatic stability, and corrosion resistance common to any marine buoy type system as well as controllability, cost and availability, etc., had to be addressed.

The system design also had to include a means for getting the wave follower from the deck of the R/V Acania into the water, positioning the

container and float and then retrieving them. Neither the present hoisting gear aboard the Acania was suitable for this nor was the boom used to position and retrieve the present float (due to increased weight) satisfactory. For this reason a derrick was designed to perform these tasks. The details of this design are found in Daley [5].

IV. HYDRAULIC DESIGN

A. SYSTEM LAYOUT

The major components of the system are:

the container (oil reservoir)
electric motor
gear pump
relief valve
filter
check valve
electrohydraulic servovalve
double rod piston

Figure 3 illustrates how system elements are arranged. The planning of the overall layout was driven by determinations involving hydrostatic stability, heat dissipation and hydraulic power considerations.

To achieve hydrostatic stability the system had to distribute weight evenly about the system center. The two components which had the greatest total weight, excluding the container itslef, were the pump electric motor combination and the hydraulic oil in the reservoir. Therefore, it was necessary to place these two items such that they balanced each other.

Hydraulic reservoirs typically contain between 2.5 and 3.5 times the pump rating in gallons per minute. The system pump is a 2.5 GPM unit. Thus, the reservoir was sized to hold up to 8.0 gallons of oil. This translated to a weight of approximately 55.0 LB. The pump motor combination weighs 52.0 LB. (dry weight). Thus, the oil weight and pump motor weight were nearly equal and were placed at opposite ends of the container. Since there was no unique amount of oil to be stored in the reservoir, and the reservoir was somewhat oversized, the oil load could be varied up to a point to trim the system for hydrostatic stability.

The remaining components were also arranged to provide stability. The filter and relief valve were located on one side of center, and the servovalve was located opposite them. The check valve, of negligible weight, was located approximately at system center. From an hydraulics viewpoint this system configuration was good, although from a heat dissipation viewpoint it was not optimal.

Suction was taken from a central point in the reservoir through a cone shaped fitting to avoid turbulence at the inlet. At the base of the cone was a 100 mesh screen which was used to protect the pump. The oil was discharged through an orifice with a 1/4 IN I.D.; 1/4 IN I.D. lines were used throughout the entire system. After the reduction, there was a pressure sensor monitoring pump output. Next was the relief valve set at a pressure to provide an adequate value at the servovalve and the piston (See section on System Pressure). The relief valve had a return line to the oil reservoir. The oil was then filtered and flowed through a check valve. The placement of the check valve insured that contaminants were not flushed back toward the pump if there were a surge opposite to the normal flow direction. Filtration requirements for the servovalve were such that the filter was placed directly up stream of it. Oil entered the pressure side of the servovalve and was either sent to one side of the piston or returned to the reservoir. This completed the oil flow path.

B. CYLINDER SIZING

The selection of the cylinder was made subject to several constraints. First, the cylinder had to be a double rod to allow easy attachments of instruments over the full rod length. A double rod design also decreases bearing loads on the cylinder cap under any transverse loading. The

cylinder had to be an off the shelf item for procurement and replacement parts considerations. Projected area was to be kept to a minimum while allowing a cylinder strong enough to withstand loading. The final consideration was the need for low fluid velocity in the cylinder and through the inlet ports.

The system was designed for a payload of 25.0 pounds. The smallest available cylinder was one with a 1.0 IN bore and a 5/8 IN rod. The weight of the rod plus the weight of the entrapped oil also had to be considered.

Then the total weight to be moved, W_T, was:

$$W_T = W_P + W_R + W_O$$

W_P = 25.0 LB (design parameter)

W_R = (rod area) (rod length) (specific weight of rod) = 7.83 LB

W_O = (annular area of oil)(cylinder length)(specific weight of oil)

= 0.72 LB

W_T = 33.55 LB

Using a load safety factor of 2.0

$WT' = (W_T)\ (S.F.) = 67.10$ LB

The maximum acceleration that the system would be called upon to respond to was assumed to be that of gravity. Thus, the thrust the system had to accommodate was

$$T = m_t\bar{A}_p + W_T'$$

$$= 134.20 \text{ LB}$$

The pressure inside the cylinder was that calculated to determine whether or not a standard or thick wall cylinder was necessary. Thus

$$P = F/A = T/A$$

$$= 280.4 \text{ PSI}$$

Using data derived from system parameters in conjunction with off the shelf items, a medium duty cylinder rated at 1000 PSI with a 1.0 IN bore and 5/8 IN rod was determined to be sufficient. To decrease further the possibility of damage due to high bearing stresses at the cylinder head a two inch stop tube was used at each end. Addition of the stop tube increased the overall stroke length from 40.0 to 42.0 IN. Further protection of the cylinder was achieved by adding an adjustable cushion head at each end.

An additional check using the Piston Rod - Stroke Selection Graph (Figure 4) was made. A stroke of 42 inches with a thrust of 134.2 LB yielded a rod of 5/8 IN diameter.

C. SYSTEM PRESSURE

The thrust load requirement was determined to be 134.20 LB and the system configuration had been chosen. The losses through the system components and piping along with the required load pressure were then summed to determine the necessary pump discharge pressure.

1. Component Pressure Drop

Each component is listed along with its contribution to pressure loss as a function of K (loss factor) in Table II.

TABLE II: Loss Factors

Component Name	K Value	Number in System	Total K
Elbow	30	3	90
Relief Valve	420	1	420
Check Valve	2000	1	2000
Line (70 inches)	(length)(I.D.)	-	.123
			2510.123

Pressure drop through the eight system orifices was taken into account by adding 1.0 for each orifice to the total K value. Summing the K^s for the system:

$$K_T = \Sigma K^s = 2510.123 \quad (f) + 8 , \qquad = .025$$

$$= 70.75$$

The pressure head resulting from these losses is, according to Crane [4]

$$h_L = \frac{.00259 \, KQ^2}{d^4} = 46.91 \, Q^2$$

There is an elevation head loss and a pressure drop across the system filter of 15 PSI in the worst case. These factors were added to the head loss calculations above. The result is shown graphically as Figure 5.

Employing the Bernoulli equation, the system pressure loss excluding the servovalve was determined.

$$\Delta P = \frac{\rho}{144} \left[(\Delta Z) + h_L \right] + \Delta P_{filter}$$

At maximum flow rate

$$\Delta P = 245.71 \text{ PSI}$$

2. Servovalve Pressure Drop

Servovalve pressure drop P_V is 200.0 PSI. The system supply pressure can be determined by the relationship:

$$P_S = P_V + P_R + P_L \, , \quad P_R = 0 \quad P_L = 280.4 \text{ LB}$$

$$= 480.40 \text{ PSI}$$

3. <u>System Supply Pressure</u>

Taking into account the losses through the system of 245.71 PSI the required pump discharge pressure is 726.11 PSI. The relief valve was set to limit maximum, system pressure to 730 PSI.

D. SYSTEM FLOW

Two aspects of system flow had to be considered. First, the volumetric flow had to be sufficient to perform the designated tasks. Secondly, the maximum flow velocity had to be kept below 15.0 FT/SEC to avoid excessive losses and unwanted pressure surges.

1. <u>Flow Volume Requirement</u>

The maximum flow requirement was fixed by the cylinder's annular area and the stroke length and velocity. The annular area was determined to be 0.48 IN^2. The maximum velocity of the cylinder velocity (at 1.0 HZ) was 17.58 IN/SEC. Thus, the maximum flow rate required at the piston was 8.44 IN^3/SEC or 2.19 GPM.

2. <u>Flow Velocity Constraint</u>

The minimum area for flow was that through the cylinder ports and lines; each has an inside diameter of 1/4 IN. Thus, the maximum velocity was determined by:

$$V_{max} = Q_{max}/A_{min}$$
$$= 14.33 \text{ FT/SEC}$$

Therefore, the maximum fluid velocity would remain below the limit of 15.0 FT/SEC.

E. SELECTION OF MAJOR SYSTEM COMPONENTS

1. Pump Motor

The system pressure and flow requirements having been determined a pump and motor combination had to be selected. A primary consideration in this selection was the type and rating of power available to the electric motor. The power source for the wave follower system was to be the R/V Acania. The Acania has 110V, 220V and 440V single and three phase A.C. power available. Likewise weight and size of the pump motor combination had to be considered.

Various manufacturers' catalogues were checked, and the pump and motor were selected from shelf inventories. The electric motor chosen was a 220/440V, three phase induction motor rated at 2 H. P. This motor was to be used in conjunction with a positive displacement gear pump. The pump driven by the 2 H. P. motor had a maximum flow capacity of 2.5 GPM at 1200 PSI. These values were commensurate with the prescribed system pressure and maximum flow rate. This eliminated the need for an intensifier. The motor pump combiation was also satisfactory from a size and weight viewpoint. The construction of these two items eliminated the need for the design of a shaft/coupling arrangement.

2. Relief Valve

The relief valve had to be able to prevnet system pressure surges which could adversely effect components. It had to be able also to operate at the flow rate of the pump selected as an upper bound. These two criteria fixed the valve selected. The valve chosen "sat" on the system in a T connection and dumped the fluid if pressure was above the predetermined system pressure.

3. Filter

The filter was located just upstream of the check and servovalves. The servovalve used required filtration of 10μ nominal and 40μ absolute. The filter had to be compatible also with system flow of 2.12 GPM and maximum pressure of 730 PSI.

The filter chosen was a MOOG type HP-10 with a filtration of 10μ nominal and 30μ absolute rated at 10 GPM and a collapse pressure of 3000 PSID. Using a 10 GPM filter in the system with maximum flow of 2.12 GPM provided for lower maintenance and greater contaminant removal. Additionally, a 100 mesh screen was provided at the pump inlet, and pump suction was taken from midpoint in the reservoir to avoid introduction of silt which would settle to the reservoir bottom.

4. Check Valve

A check valve was used to prevent back flow through the filter. Backflow would reintroduce particulate contaminants into the system. The check valve selected was an inline type with 1/4 IN I. D. (consistent with that of the lines) with a maximum pressure rating of 3000 PSI.

5. Electrohydraulic Servovalve

This valve was the heart of the system. The valve selected had to be able to operate at system pressure and flow rates. The valve chosen was a critical center, full periphery ported, four way servovalve, the MOOG 62-105. The critical center valve was chosen over either the overlapped or underlapped valve due to inherent drawbacks of these other types. The underlapped valve has a large power loss, and the overlapped valve has a deadband which results in severe position error when used in a VCP system.

The 62-105 servovalve at flow rate of 2.5 GPM had a valve pressure loss of 200 PSI. The valve had a standard maximum pressure rating of 2000 PSI.

In service the servovalve used is to be supplied with a dither signal of 85 HZ at 5 MA peak as recommended by the manufacturer. This would prevent silting of contaminants which could lead to poor system behavior and degrade control at low speed.

6. <u>Hose and Fittings</u>

The hose was selected to be compatible with system pressure, hydraulic oil, and expected maximum oil temperature and of a size to limit maximum flow velocity to the 15.0 FPS maximum. The hose also was selected with a minimum bend radius criteria compatible with system size and layout.

Fittings were of the same size and thread design to facilitate replacement. All system components, e.g. filter and servovalve, were fitted with inserts so that they too would have the same size and thread type as the hose fittings. The hose and fittings were of the "no skive" type for each of fabrication and replacement.

7. <u>Hydraulic Oil</u>

The oil selected was CHEVRON No. (11) 46 paraffin base hydraulic oil. Specifications are contained in Table III. The oil was designed for use in hydraulic systems of medium pressure, and it contained corrosion inhibitors along with foam and aeration suppressants. A high viscosity index oil results in small viscosity changes with temperature. The oil was recommended for use with vane, piston, and gear pumps.

TABLE III: Hydraulic Oil Parameters

Base: Paraffin

Grade Designation: 46

Viscosity			
100°F	210°F	100°F	210°F
215 SUS	48 SUS	46 CST	6.7 CST

Viscosity Index: 95

Flash Point : 425°F

Pour Point : -20°F

Aniline Point : 218°F

Additives : Foam and Aeration Suppressants
Corrosion Inhibitors

V. CONTAINER DESIGN

A. CONTAINER SIZE AND SHAPE

The container was designed to fit an existing float. The height of the container was fixed by size restrictions of the components. There had to be sufficient space between components to preclude the use of special tools and to allow for prescribed minimum bend radii for hydraulic hose. Within limitations heat generating components were to be separated by some distance and mounted off the container bottom. Final container size and configuration are illustrated in Figure 6. Component location is as described in Chapter IV on Hydraulic Design.

B. CYLINDER SUPPORT DESIGN

The wave follower system was to be added to an existing float. The float configuration provided a 1 1/2 IN pipe running through the float and wave follower center. The piston was to be attached to this pipe by two 30 IN arms fixed to the top and bottom of the cylinder. This mounting arrangement is allustrated in Figure 7.

The two arms were 30 IN 2 x 2 x 1/4 IN equal leg angle iron. These arms were welded to attaching fixtures at the vertical support ends and the cylinder were bolted to the opposite end. Drawings of the arm attachments are shown in Figure 8.

The maximum (bending) stresses occurred at the weld of the arms to the attaching fixtures. AISC codes were used to determine maximum allowable stresses at these butt welds. The design safety factor used resulted

in each arm's having the capacity to individually support the maximum thrust load for cylinder and payload at peak accelerations. The materials and welding rod selected were compatible with this loading.

Deflection calculations revealed that the maximum deflection at the cylinder under worst case loading would be 0.1 IN. This deflection was satisfactory. The support attachment and arms were to be coated with the same corrosion inhibiting paint as the container.

C. RESERVOIR DESIGN

A specific section on reservoir design is included due to the variety of considerations addressed. The reservoir was located at what in operation would be the forward face of the wave follower. This location afforded the greatest flow of cool ocean water thereby enhancing heat transfer through the walls.

The capacity of the reservoir, as previously mentioned, was approximately 3.0 times the pump capacity. Therefore, its dimensions were bounded.

Two return lines fed into the reservoir-one from the relief valve and the second from the servovalve return port. These lines ran to opposite corners of the reservoir and were directed against the front wall, the coolest part section of the reservoir. Suction was located at approximately the midpoint, thereby decreasing the chance for ingesting either silt, which settled to the reservoir bottom, or any floating particulate contamination.

Baffles were provided to minimize any rapid oil motion due to container motion which might cause oil foaming and adversely affect the oil's bulk modulus. The baffles were constructed from 1/4 IN perforated mild

steel. A breather cap was located at the tank top center to avoid reservoir pressure build up due to oil heating. A drain was installed below the level of the return lines.

D. MATERIALS SELECTION

Primary concern in the materials selection was the ability to fabricate the design with existing machines, techniques and on-hand personnel. This concern was followed by the problem of material strength, availability, price and specific weight. Use of the system in a marine environment would promote corrosion, and the material also had to be judged on its heat transfer/dissipation qualities.

There were four basic materials from which to choose-mild steel, stainless steel, aluminum and plexiglass. Mild steel was the least expensive of the above materials; it had sufficient strength, was easy to fabricate, had fair heat transfer properties (although not as good as aluminum, but better than plexiglass), and was readily available. However, it has fairly high specific weight for commensurate strength compared with the other materials. At this point, mild steel was preferable.

Corrosion problems would result not only from the marine environment but also from galvanic attack due to dissimilar metals in contact. The motor was mild steel; the servovalve and filter were aluminum. In this vein, plexiglass would be the best base material. However, other properties, e.g. heat transfer, overruled its use. Consulting the electrolytic chart revealed that the minimum potential (for the metals) would dictate either mild steel or aluminum. Stainless steel had the drawback of either being active or passive which would lead to rapid deterioration.

There was a large variety of paints easily procurable and applicable that would protect any of these materials from the marine corrosion

problem. Therefore, any of these metals could be protected against rapid sea salt corrosion.

A synthesis of the information led to mild steel as the choice for container material. Mild steel was inexpensive, available, and easily fabricated; it had good strength to weight and rigidity characteristics, fair heat transfer properties, and good galvanic series location for minimizing attack.

E. WATERTIGHT INTEGRITY

The wave follower container had to be water tight to insure component operation. One large piece of steel was used to form the base and sides, thus eliminating the need to make two 40.0 IN welds. The ends were also bent into channel like shapes and fitted over the main piece. These ends were butt welded to the main section lap welded along the channel ends to the sides of the main section, thus providing double weld protection at all four edges.

Along the inside at the top 1.0 IN equal angle iron was welded in. This angle iron was drilled and topped every 3.0 IN and covered with 1/4 IN thick rubber to provide the water tight seal. A 2.0 IN pipe passed through the container to allow the cylinder vertical support to be attached to the float. The pipe was welded into the container bottom. The pipe was machined at the top and fitted with an O ring seal. The container top had a hole with a removable flange that fitted over the O ring.

Tests were run on the container to determine if there were any leaks into the container either from the otuside or from the oil reservoir into the rest of the container. Fortunately, no leaks were discovered.

F. HEAT DISSIPATION

The electric motor and servovlave would be the largest source of heat generation. In an effort to ameliorate the heat buildup all components were mounted 1/2 IN above container bottom, thereby allowing air flow on all sides. The motor was equipped with a fan to promote air circulation. The container will be partially submerged during operation, thus providing a heat sink through container walls.

The possibility of using a submersed electric motor, as discussed below, was abandoned due to the cost factor involved. In an attempt to overcome this problem, tests were conducted with a small horsepower induction motor submerged in an oil bath. The motor was run with no load to determine the feasibility of using an off the shelf motor inside the oil reservoir. The motor was run for eight hours on several occasions. Data were gathered as to oil temperature versus time of operation and on each occasion the oil temperature did reach a steady state.

Although these tests indicated that this idea was feasible, there were too many unknowns (e.g. Would hot oil attack motor insulation, how much motor horsepower would be consumed in operation in this viscous environment and would ingestion of silt into the motor promote rapid wear?). Additionally, the problems of maintaining hydrostatic equilibrium would be compounded. It was felt that the risks outweighed the possible benefits, and this concept was abandoned.

G. COMPONENT SELECTION TRADE-OFFS

The wave follower was designed to respond to those frequencies and amplitudes as discussed in the design specifications. Component selections were driven by these requirements. As the system began to take form and component selection began, trade-offs also began.

System weight and size were to be kept to a minimum for reasons previously outlined. The size was limited by outward dimensions of the float. Weight, the next concern, was most significantly effected by the pump motor selection and by the reservoir capacity. Selecting the cylinder, commercially available, with the minimum annular area was the first step toward keeping flow volume small.

Having selected the cylinder and working toward system requirements, Table I was prepared. Working from these data, pump motor selection began. The first choice was a combination that provided volume flow of 5 GPM. This, in conjunction with needed system pressure, led to the motor size selected. The combination originally selected was a 5 GPM pump with a 9 H. P. electric motor.

This pump motor set would fulfill system requirements, but the total weight of these components was unacceptable. Working with manufacturers' catalogues, the iterative process toward a final selection was pursued.

Low frequency response was sacrificed in this selection. However, judicious selection of the configuration and the amount of reserve buoyancy in the system float provided a platform for the wave follower which itself responded to these low frequency disturbances.

Other areas for trade-offs concerned price, component availability, and heat dissipation. Again the pump motor component selection is used as an example. A larger pump motor combination was inherently more expensive. Additionally, when a 9 H. P. motor and the selected 2 H. P. motor (operating at the same efficiency) were compared for heat generation, clearly the 2 H. P. motor created less heat. In an enclosed container this would be of significant importance.

In an attempt to alleviate this problem, inquiries were made as to the availability of an off the shelf motor of sufficient rating that could be immersed in the reservoir. The results were that this submersible motor would cost up to 6 times that of the motor selected and possessed a delivery date of several months. Both of these factors were unacceptable.

One final example of the trade-offs encountered was that of a sensor used to determine cylinder position. The original desire was to use a LVDT (linear position transducer). However, the LVDT required a high frequency voltage source. The other alternative was the use of a pulley arrangement with a multi-turn potentiometer. This alternative was selected since it could be accomplished at minimal cost and with the materials available on hand.

VI. CONTROLS DESIGN

A. OPEN LOOP SYSTEM

The combination of an electrohydraulic servovalve and a piston was the heart of the wave follower system. Three specifics of this system, the transfer function, the hydraulic damping, δ_h, and the hydraulic natural frequency, ω_h, were of paramount importance.

The transfer function for a VCP system in the S domain is

$$X_p = \frac{(K_q/A_p)X_v - (K_{ce}/A_p^2)(1+(V_t/4\beta_e K_{ce})-S)F_L}{\frac{V_t M_T}{4\beta_e A_p^2}S^3 + [\frac{K_{ce}M_t}{A_p^2} + \frac{\beta_p V_t}{4\beta_e A_p^2}]S^2 + [1+\frac{B_p K_{ce}}{A_p^2} + \frac{KV_t}{4\beta_e A_p^2}]S + \frac{K_{ce}K}{A_p^2}}$$

In this system, $F_L = 0$ since no load force was applied to the piston. Additionally, no spring loads were present, and hence $K = 0$. The factor, $B_p K_{ce}/A_p^2$, is much smaller than one, $B_p \simeq 0$. Thus, the transfer function for this system became:

$$\frac{X_p}{X_v} = \frac{(K_q/A_p)}{[\frac{V_t M_t}{4\beta_e A_p^2}]S^3 + [\frac{K_{ce}M_t}{A_p^2}]S^2 + S}$$

In order to use this transfer function the valve coefficients, K_q, K_{ce}, and system constants, V_t, M_t, A_p, along with the equivalent bulk modulus, β_e, had to be determined. The valve coefficients were determined from the manufacturer's data.

Flow gain K_q is $\partial Q/\partial X_v$. However the valve would spend most of its time centered at $X_v = 0$ and $P_L = 0$. Thus, the salient K_q valve would be the null flow gain $K_{qo} = \partial \underset{\sim}{Q}_L/\partial X_{vo}$. K_{qo} was evaluated in two ways.

From valve curve determinations, it is found that

$$K_q = c_d W \sqrt{\left(\frac{1}{\rho}\right)(P_S - P_L)}\ , \qquad P_L = 0$$

$$K_{qo} = \frac{c_d}{\sqrt{\rho}} W \sqrt{P_S}$$

Here, W is the valve gradient. The valve gradient, as stated by the manufacturer, was 0.082 IN^2/IN. Then

$$K_{qo} = 83.2\ IN^3/SEC/IN$$

The flow gain also was obtained from flow input current (torque motor) information. The flow gain was 0.05 GPM/MA for full deflection I_{max} = 100 MA. Thus, maximum flow was 5 GPM. The maximum input current corresponded to a full deflection of the valve spool of 0.1 IN. Converting 5 GPM to 19.25 IN^3/SEC led to a flow gain value of 192.5 $IN^3/SEC/IN$. This value is predicated on a valve drop of 1000 PSI. In this system the valve pressure drop is 200 PSI. Correcting the flow gain for this factor gave

$$K_{qo} = 192.5 \sqrt{\frac{200}{1000}}$$

$$K_{qo} = 86.1\ IN^3/SEC/IN$$

The slight disparity in the two evaluations of K_{qo} resulted from a difference in the oil density value incorporated in the manufacturer's data and that utilized in this system analysis.

The null pressure coefficient K_{co} is $\partial QL/\partial P_{LO}$. It was evaluated as

$$K_c = \frac{c_d W X_v \sqrt{(\frac{1}{\rho})(P_s - P_L)}}{2(P_S - P_L)}, \quad P_L = 0$$

$$K_c = 0.0203 \text{ IN}^3\text{/SEC/PSI}$$

It must be pointed out that K_{co} was used in the VCP transfer function and not K_{ce} ($K_{ce} = K_{co}$ + leakage flows). The small amount of system leakage flow was assumed to be negligible.

The pre multiplier of the S^3 term in the transfer function is defined as the square of the reciprocal of hydraulic natural frequency ω_h. That is

$$\omega = \sqrt{\frac{4\beta_e A_p^{\,2}}{M_t V_t}}$$

A nominal value for bulk modulus (β_e = 100,000.0) was used in all calculations. The values for M_t and V_t are determined in other sections of this paper. Inserting the appropriate values yields a hydraulic natural frequency of

$$\omega_h = 218.82 \text{ RAD/SEC}$$

The coefficient of the S^2 term in the transfer function is defined as twice the hydraulic damping divided by the hydraulic natural frequency (i.e., $2\delta h/\omega_h$). The hydraulic damping is defined as

$$\delta_h = \frac{K_c}{A_p}\sqrt{\frac{\beta_e M_t}{V_t}} = 0.708$$

This value of δ_h appears to be good. However, it must be recalled that certain assumptions were made during the development of flow

coefficients (e.g., $c_{tp\ell} = 0.0$); thus, the actual δ_h value would be somewhat different than that calculated.

Having determined the necessary constants the transfer function for the valve controlled piston became

$$\frac{X_P}{X_v} = \frac{(83.2/0.4786)}{S\left[\frac{S^2}{(218.82)^2} + 2\,\frac{0.708}{(218.82)}\,S + 1\right]}$$

$$= \frac{173.84}{S[2.09\text{x}10^{-5}S^2 + 6.47\text{x}10^{-3}S+1]}$$

The system poles were:

r_1: $S = 0.0$

r_2: $S = -154.75 + i\ 154.57$

r_3: $S = -154.75 - i\ 154.57$

The system, as it stands with a real root to the right of the two imaginary roots, will be sluggish. However, this open-loop analysis and transfer function development did not take into account the feedback mechanism. The overall system transfer function is shown in the next section.

B. CLOSED LOOP SYSTEM

The feedback and control portion of the wave follower system could be broken into four main components. These components were the power supply, the wave height sensor, the cylinder position indicator and the summing amplifier. The role of each component in the control scheme will

be discussed in this chapter. Detailed electronic operation and schematics are covered in a Naval Postgraduate School Technical Note (16).

The wave height sensor or wave gauge provided an electrical signal, e_{WG}, proportional to wave height. The cylinder position indicator provided a signal, e_{CPI}, defining cylinder position. These two signals were fed into a summing amplifier and compared. The difference in amplitude was the error between the wave position and the cylinder position. The error signal was fed to the torque motor of the servovalve, either increasing or decreasing oil flow to the cylinder and thus positioning the cylinder.

Analysis and design of feedback and control systems can be done in a myriad of ways. One particularly powerful tool is the use of state variable analysis in order to generate a transfer function in the S domain. This is the method that was employed in this study.

The wave follower system was a valve controlled piston (VCP) circuit. The transfer function and state variable matrices for this system are well known. The development can be found in Houlihan [10]. The transfer function for a VCP is:

$$\frac{X_P}{X_v} = \frac{K_{qo}\ A_p}{\left(\frac{1}{\omega_h}\right)^2 S^3 + 2\frac{\delta_h}{\omega_h} S^2 + S}$$

The state variable matrix representation of the open loop system, Merritt [13], is shown below.

$$\begin{bmatrix} P_L \\ X_p \\ \dot{X}_p \end{bmatrix} \begin{bmatrix} S + (4\beta_e K_{ce}/V_t) & 0 & (4A_p/\beta_e V_t) \\ 0 & S & -1 \\ -(A_p/M_t) & K_p\ M_t & (S + B_p/M_t) \end{bmatrix}^{-1} \begin{bmatrix} \\ X \\ \\ \end{bmatrix}$$

$$+ \begin{bmatrix} (4\beta_e K_q/V_t) & 0 \\ 0 & 0 \\ 0 & -(VM_t) \end{bmatrix} \begin{bmatrix} X_V \\ \\ F_L \end{bmatrix}$$

P_L and $\dot{X}_p$ are zero since in this system only X_p (cylinder position) was used as a feedback. System and valve parameters, along with the above mentioned assumptions, were inserted into the state variable matrices yielding

$$\begin{bmatrix} 0 \\ X_p \\ 0 \end{bmatrix} \begin{bmatrix} S + 312.82 & 0 & 7303.25 \\ 0 & S & -1 \\ -6.56 & 152.60 & S \end{bmatrix}^{-1} \begin{bmatrix} \\ X \\ \\ \end{bmatrix}$$

$$+ \begin{bmatrix} 1.27\text{x}10^6 & 0 \\ 0 & 0 \\ 0 & -13.70 \end{bmatrix} \begin{bmatrix} X_v \\ \\ 0 \end{bmatrix}$$

Up to this point the system was open loop.

Position feedback was now inserted using the cylinder position indicator with a gain of K_s. The gain was determined simply by dividing full scale voltage by full scale cylinder travel.

$$K_s = \frac{\text{Full Scale Voltage}}{\text{Full Scale Cylinder Deflection}}$$

$$= 0.27 \text{ V/IN}$$

The wave gauge signal was converted to an electrical output in the wave gauge electronics. The wave gauge was modeled as a unity gain amplifier, Kw, since its gain would be tuned to give a one to one correspondence in volts per inch with the cylinder position feedback network. The signal was compared with the e_{WG} signal in a variable gain op-amp. The upper bound was determined by system stability considerations to be 6.73. The amplifier gain was designated Ka in the block diagram.

Error voltage was then fed into the torque motor which positioned the valve spool. The torque motor was modeled as a simple lag system with unity gain $\left(\frac{\text{inches spool travel}}{\text{volt}}\right)$. This assumption is normally valid, and especially so in the low frequency range. The range of interest for this design was 1.0 HZ to 10 HZ. The torque motor positioned the valve spool; its deplacement was X_v. The direction and amount of spool displacement was controlled by the error signal circuit which acted to minimize the displacement error. The closed loop block diagram of the system is shown as Figure 9. Manufacturer's data was used to determine the time constant of the torque motor (τ_c = 0.032 SEC).

The system above was reduced to yield the closed loop system transfer function. The transfer function is ($\underline{\bar{K}} = K_a\ K(K_q/A_p)$)

$$\frac{X_W}{X_p} = \frac{\underline{\bar{K}}\ K_s}{[\tau(\frac{1}{\omega_h})^2]S^4 + [\frac{2\tau_c\delta_h}{\omega_h} + (\frac{1}{\omega_h})^2]S^3 + [\frac{2\delta_h}{\omega_h} + \tau_c]S^2 + S + \underline{\bar{K}}\ K_s}$$

The closed loop transfer function with the inserted constants (amplifier gain Ka = 1.0) is

$$\frac{X_W}{X_p} = \frac{46.19}{[6.68\text{x}10^{-7}]S^4 + [2.29\text{x}10^{-4}]S^3 + [3.85\text{x}10^{-2}]S^2 + S + 46.19}$$

C. SYSTEM MODELING

The system's closed loop response was modeled on the IBM/360 using the Continuous System Modeling Program - CSMP - Speckhart and Green [22]. The closed loop transfer function coefficients were loaded into the predefined CSMP transfer function routine. A SINE wave, varied from 1.0 HZ to 10.0 HZ in increments of 1.0 HZ, was used as system input. The output, piston position, X_p; and driving function were plotted at each of the ten forcing frequencies, Figures 10-19. These results show the phase relationship between the error signal, e_s, and piston position, X_p. A system BODE diagram, Figure 20, was also plotted. The BODE diagram approximated the classic second order curves due to the small S^4 and S^3 coefficients. There was a steep drop in the amplitude ratio above 6.0 HZ. However, since the maximum height of the waves decrease with increasing frequency, this response is satisfactory. The computer programs that generated Figures 10-20 are contained in Appendix B.

D. SYSTEM ELECTRONICS

The electronics package consists of five elements:

Servo Amplifier
Wave Gauge
Cylinder Position Indicator
Window Comparator
Dither Signal Generator

With the exception of the first item all other components were designed and constructed by Mr. T. Christian, electronics technician for the Department of Mechanical Engineering.

1. Servo Amplifier

The servoamplifier was the SNAP TRAC SERVO ELECTRONICS manufactured by MOOG, INC. It supplied the necessary power to operate the servovalve's torque motor and served as the source of power for the cylinder position indicator. In addition, the servoamplifier contained the op-amp used in generating the error signal as well as the amplifier shown as K_a in the block diagram.

2. Wave Gauge

The wave gauge was basically a capacitance bridge network. A copper wire was fixed to the wave follower at the top and was secured to the float in the water. As the water height changed, the capacitance in that leg varied. This signal was fed back through the electronics and was converted to a voltage which signified the water's height.

3. Cylinder Position Indicator

The indicator was a ten turn precision wire wound potentiometer. A wheel was fixed to the shaft with a small cable running one turn around the wheel. The cable was fixed to the top and bottom of the piston rod. As the piston moved inside the cylinder body the cable turned the wheel varying the resistance. Thus, the voltage feedback to the summing amplifier was a linear function of cylinder position.

The cylinder position indicator and the wave gauge were located in an aluminum housing at the top of the cylinder. The housing arrangement for this is shown in Figure 21.

4. Window Comparator

The window comparator was located adjacent to the servoamplifier. The purpose of the comparator was to protect the cylinder from any damage that would result from an excessive error signal. If the error signal exceeded $\pm$ 4.5 V, the comparator automatically set the error signal to zero. The valve flow forces then returned the valve spool to the null position. This arrangement precluded the possibility of the system's pushing the rod against the cylinder head.

5. Dither Signal Generator

As recommended by the servovalve manufacturer, a dither signal was supplied to the valve spool. The dither signal generator was located on the same board as the window comparator. The dither signal was used to prevent silting of particulate contaminants at the valve spool lands. This signal was a sinusoidal signal of 85 HZ with a valve peak amplitude of 5 MA. Schematic diagrams of these five components and a more complete description of these circuits are contained in a Naval Postgraduate School Technical Note (16).

VII. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

An electrohydraulic wave follower system was designed and modeled. The system container, electronics and supporting attachments were constructed. The system had the following features:

1. The system response exceeded that determined by linear wave theory, using a steepness criterion of one seventh for a frequency range of 1.0 to 10.0 HZ. See Table IV.

2. The 25.0 LB design payload could be increased significantly with no structural or component changes being necessary.

3. The null point could be set as desired to allow various instrumentation heights.

4. The system was marinized, e.g., designed for hydrodynamic stability and coated with corrosion inhibiting paints, to allow for long term operation in the ocean environment.

B. RECOMMENDATIONS

1. To improve system tracking at higher frequencies a lead/lag compensation network should be incorporated into the control system. System testing must be performed prior to this design.

2. For operations with payloads above design, the system pressure must be increased. This increase in pressure may necessitate addition of an accumulator. The present design has provided space for this addition.

3. To further enhance system cleanliness, thus promoting longer trouble free operation, return line filters with a 3μ absolute rating should be incorporated. These filters should be placed in both the relief valve and servovalve return lines. Sufficient space has been provided for this in the present design.

4. A new primary float should be designed. The requirement of the into-the-wind positioning of the original float's sensor and the into-the-sea requirement of this system may prove incompatible.

TABLE IV: Required and Available System Height Response for Various Wave Frequencies

Frequency (HZ)	Height Required (IN)	System Height Available (IN)
1	8.79	10.22
2	2.20	11.45
3	.98	12.68
4	.55	15.33
5	.35	17.87
6	.24	17.49
7	.18	13.70
8	.14	10.63
9	.11	8.55
10	.09	7.27

APPENDIX A: SYSTEM COMPONENTS

COMPONENT NAME	MANUFACTURER	PART NUMBER
Double rod cylinder	Parker Hannifin	1CKC3LLS24/24Cx44
Servovalve	Moog	62-105
Servo controller	Moog	121-103
Check valve	Parker Hannifin	VCL4P
Filter	Moog	HP010CP100N
Relief valve	Parker Hannifin	705661
Pump (gear)	Parker Hannifin	P-14
Electric motor (2 H. P.)	Parker Hannifin	725498 (M-8)
Hose (system)	Parker Hannifin	221-5
Male pipe swivels	Parker Hannifin	21342-4-4
Hose (suction)	Parker Hannifin	421-12
Breather cap	Parker Hannifin	675387
Cylinder position caple	Sava-Industries	2027 5N

```
*   THIS MACRO WILL FIND THE FREQUENCY RESPONSE FROM A GIVEN TRANSFER
*   FUNCTION WITH A NUM. AND DENOM. OF NO MORE THEN FIFTH ORDER.  A BODE
*   PLOT IS PRODUCED AS OUTPUT.

*   AMPLITUDE RATIO IS IN DB, FREQ.IS IN HZ.

MACRO FASE, MAG= BODE(N,D,OMEGA)
PROCEDURAL
   X1=N(5)*(OMEGA**4)-N(3)*(OMEGA**2)+N(1)
   X2= N(6)*(OMEGA**5)-N(4)*(OMEGA**3)+N(2)*OMEGA
   X3= D(5)*(OMEGA**4)-D(3)*(OMEGA**2)+D(1)
   X4= D(6)*(OMEGA**5)-D(4)*(OMEGA**3)+D(2)*OMEGA
   MAG =(SQRT(X1**2+X2**2))/(SQRT(X3**2+X4**2))
   CONV =57.29578
   XREAL= X1*X3+X2*X4
   XIMAG= X2*X3-X1*X4
   FASE= (ATAN2(XIMAG,XREAL))*CONV
   IF (FASE) 2,2,3
   3    FASE=FASE-360.
   2    CONTINUE
ENDMAC

*    AMPLIFIER GAIN=1.00

INITIAL
 PARAMETER WMIN=0.10,WMAX=69.12
 STORAGE DENCOF(6),NUMCOF(6)
NOSORT
TABLE NUMCOF(1-6)= 46.18,5*0.0,...
      DENCOF(1-6)= 46.18,1.0,3.85E-02,2.29E-04,6.68E-07,0.0
DYNAMIC
    OMEGA=WMIN*(10.**TIME)
   PHASE,XRAY =BODE(NUMCOF,DENCOF,OMEGA)
   VALUE=20.*ALOG10(XRAY)
 FREQ=OMEGA/6.28319
TERMINAL
TIMER FINTIM=100.,OUTDEL=.001,DELT=.001,PRDEL=.001
FINISH OMEGA=WMAX
OUTPUT FREQ,VALUE
PAGE HEIGHT=5,WIDTH=8,XYPLOT
END
STOP
ENDJOB
```

```
INITIAL

* PROGRAM VARYING FREQUENCY FROM 1 HZ TO 10 HZ IN STEPS OF 1 HZ

*   AMPLIFIER GAIN = 1.0

PARAMETER OMEGA=6.28
STORAGE DENCOF(5),NUMCOF(1)
NOSCRT
TABLE DENCOF(1-5)=1.0,3.85E-02,2.29E-04,6.68E-07, 46.18,...
      NUMCOF(1-1)=46.18
6000  FORMAT('1',5X,' DELTAH = ',E12.5,5X,' OMEGAH = ',E12.5,//)
      WRITE(6,6000) DELTAH,OMEGAH
DYNAMIC
NOSCRT
 EWG=SINE(0.0,OMEGA,0.0)
 EWG1=(EWG)*5.
 XP=TRANSF(4,DENCOF,0,NUMCOF,EWG1)
TERMINAL
 TIMER FINTIM=1.0,OUTDEL=.001,DELT=.001,PRDEL=.001
LABEL WAVE FOLLOWER RESPONSE
 OUTPUT TIME(0.0,1.0),EWG1(-20.0,20.0),XP(-20.0,20.0)
PAGE HEIGHT=5,WIDTH=8,GROUP=3,XYPLOT
END
PARAMETER OMEGA=12.56
END
PARAMETER OMEGA=18.84
END
PARAMETER OMEGA=25.12
END
PARAMETER OMEGA=31.40
END
PARAMETER OMEGA=37.68
END
PARAMETER OMEGA=43.96
END
PARAMETER OMEGA=50.52
END
PARAMETER OMEGA=56.52
END
PARAMETER OMEGA=62.80
END
STOP
ENDJOB
```

APPENDIX C: SYSTEM OPERATIONS

The system, once constructed, is to be flushed to remove all possible contaminants. This must be done with the servovalve out of the system. A flushing block will be inserted for this purpose. At the completion of this flush, the system filter and inlet screen will be checked.

The servovalve is then to be inserted and the mechanical null set. This setting will depend on the desired zero point, i.e., desired height above or below air sea interface.

Next, the amplifier gain, K_a, is to be set to the desired value without exceeding the upper limit of 6.72 (stability consideration). Modeling has shown that a gain of 1.0 will yield good results for operations at frequencies above 1.0 HZ. The gain of one in the modeling routine is predicated on a maximum wave gauge signal of $\pm$ 5 V.

The system will then be run using a function generator to simulate the wave gauge signal. This step will allow variation of signal amplitude and frequency without the use of a wave tank. During this phase of testing, the electric motor housing temperature will be monitored (Three thermocouples, one every 120° are fixed to the motor housing.). This test also will allow comparison of actual system output with that predicted by computer simulation.

A wave tank will be required to carry out full system testing. The container will be partially submerged, and the wave gauge set in place. Amplitude and frequency of the tests will depend on the capability of the wave tank. During this test final adjustments will be made to the

electronics to obtain desired results. These test results will again be compared with those of the simulation. Thermocouples also have been provided in the oil reservoir to monitor the bulk temperature during testing.

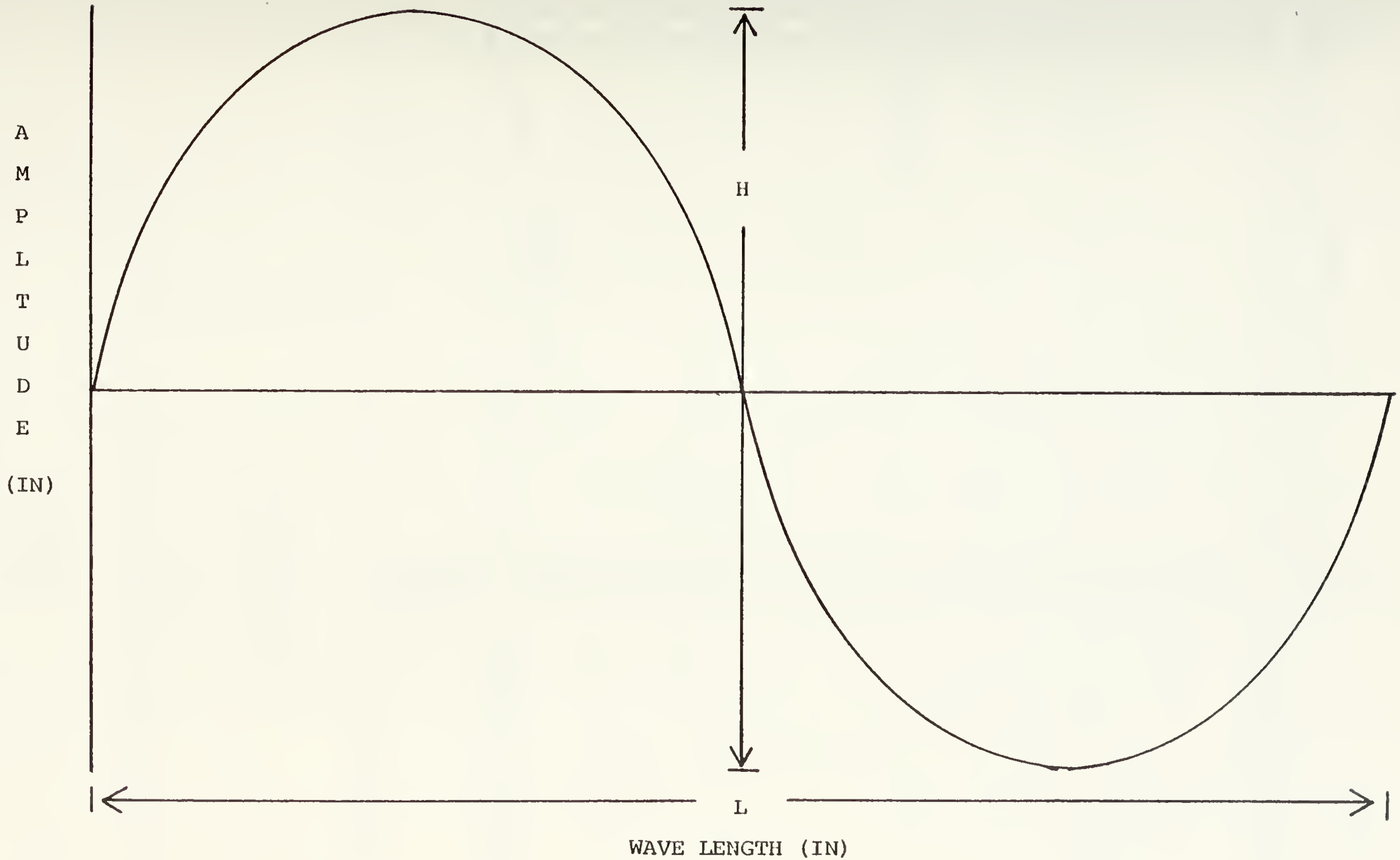

Figure 1. Wave Height and Period Description

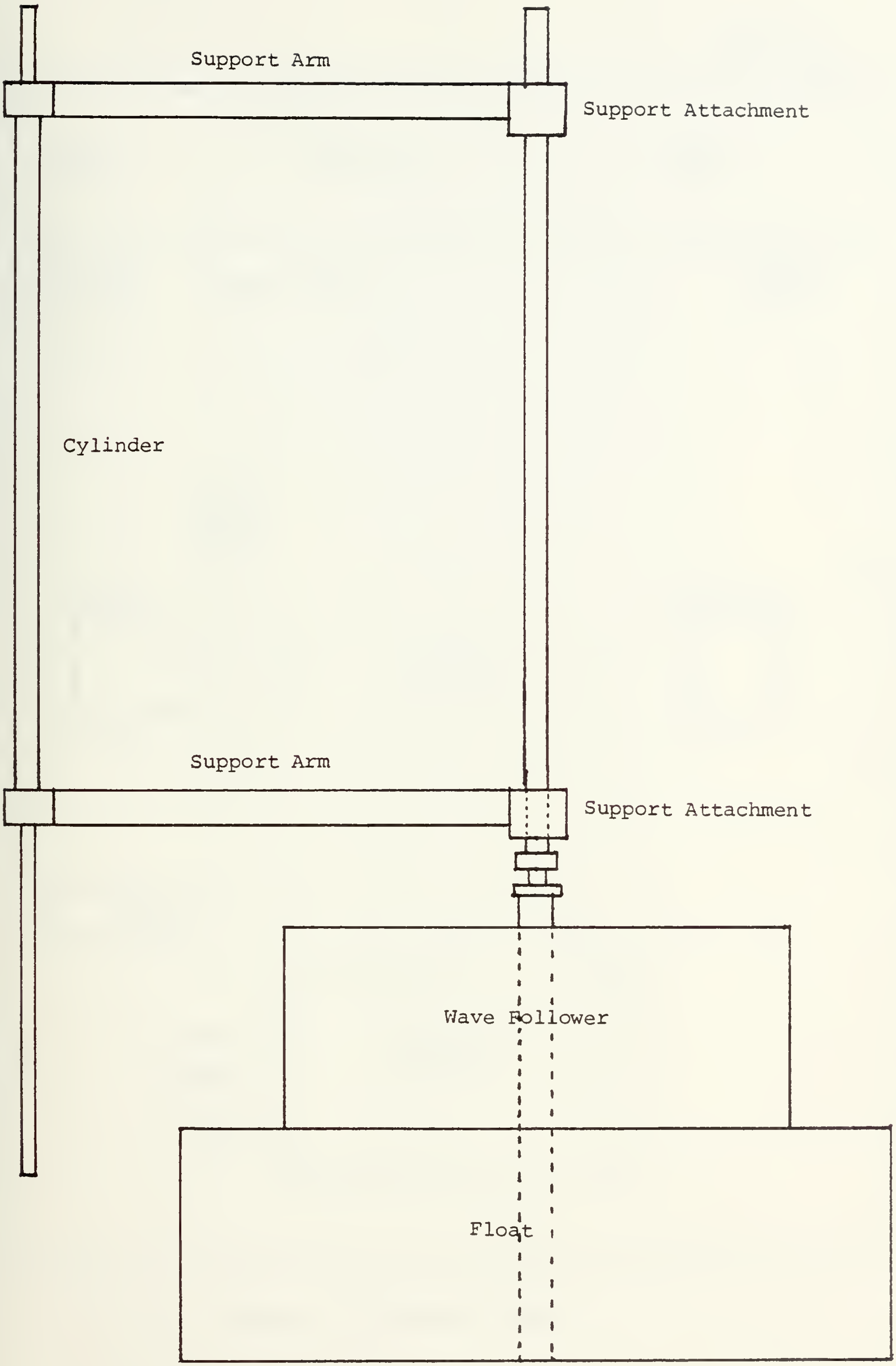

Figure 2. Wave Follower and Float.

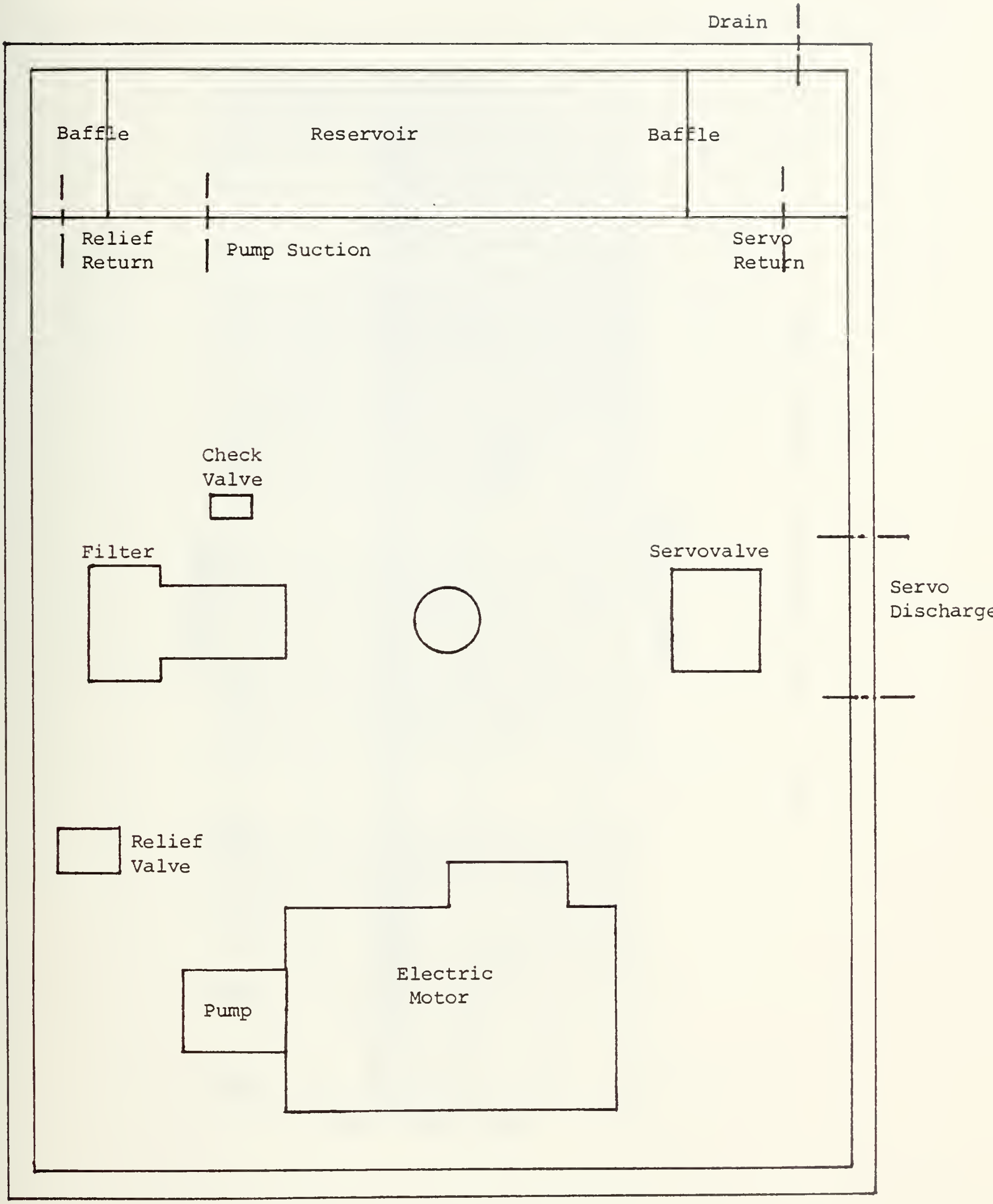

Figure 3. Component Layout.

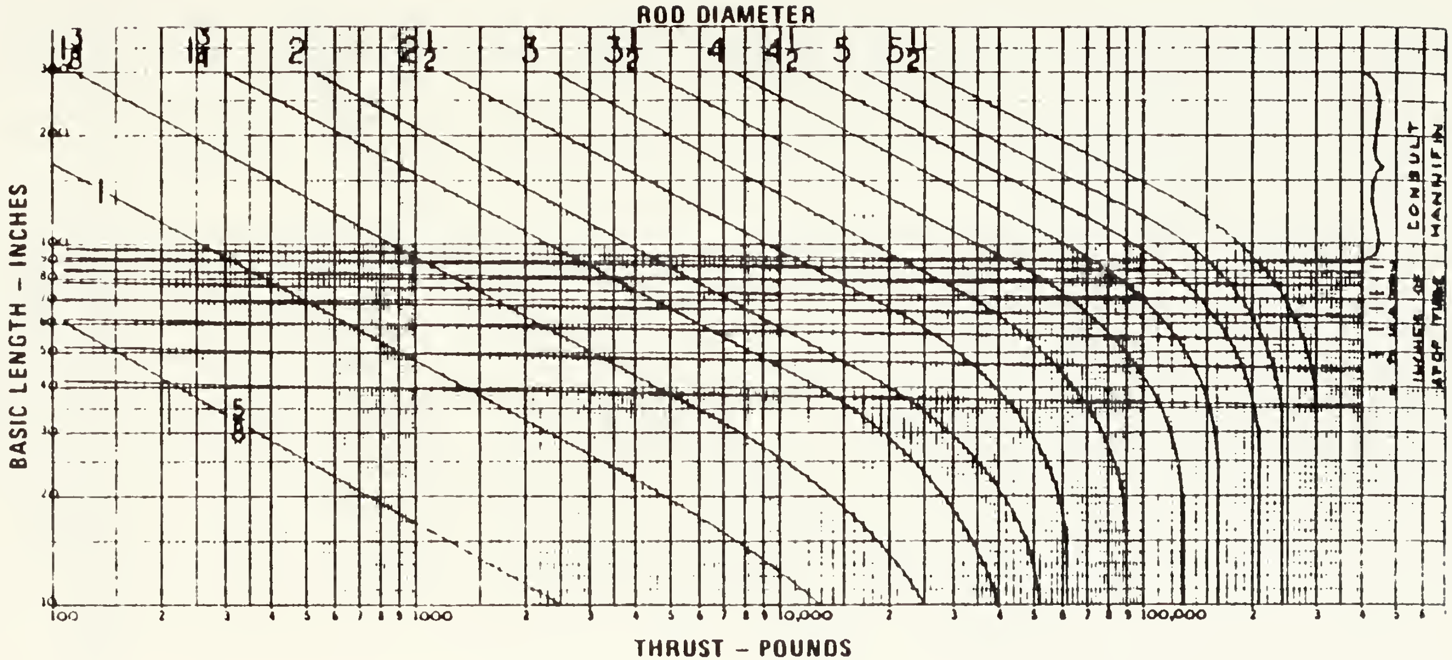

Figure 4. Piston Rod-Stroke Selection Graph.

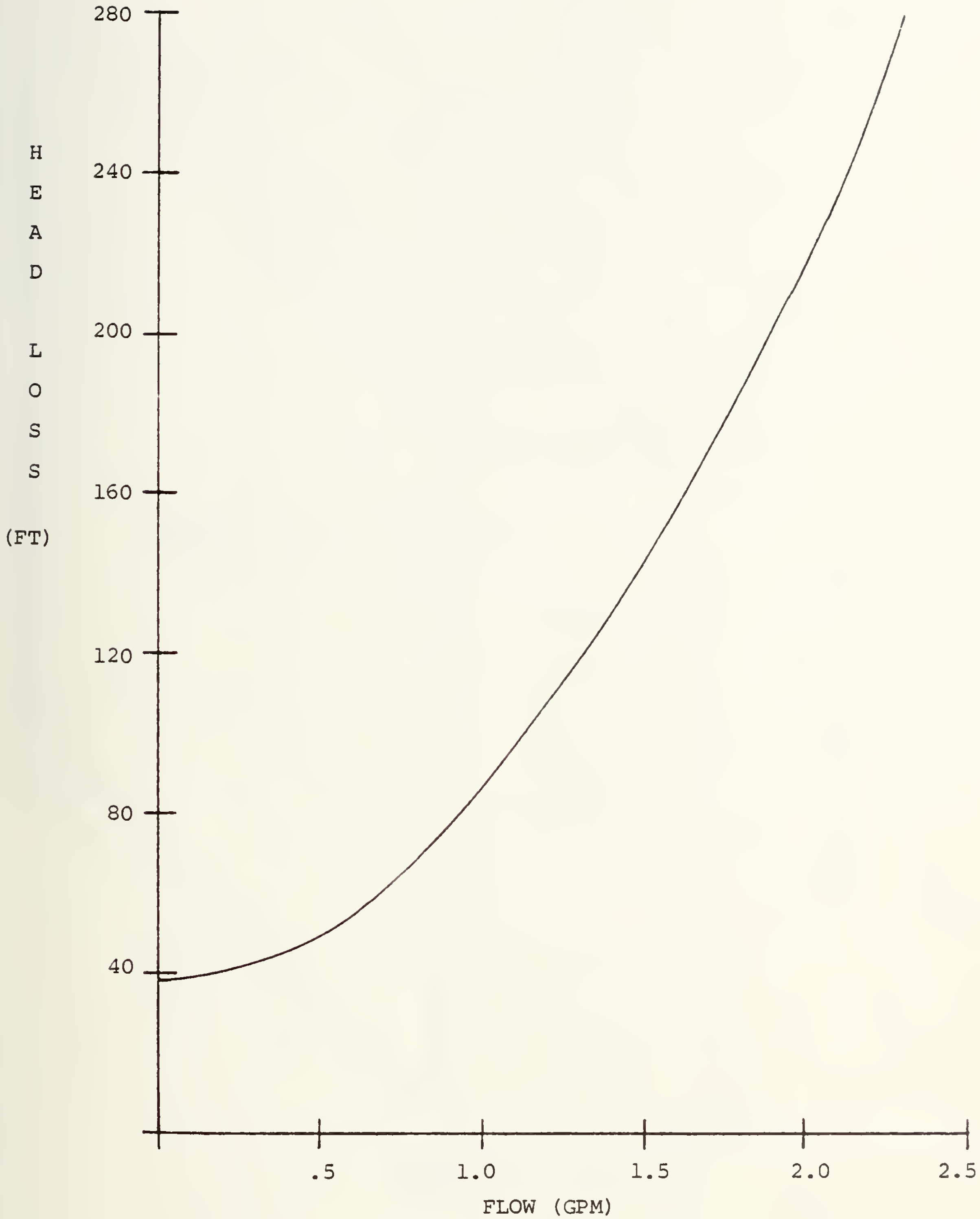

Figure 5. Head Loss-Flow Plot.

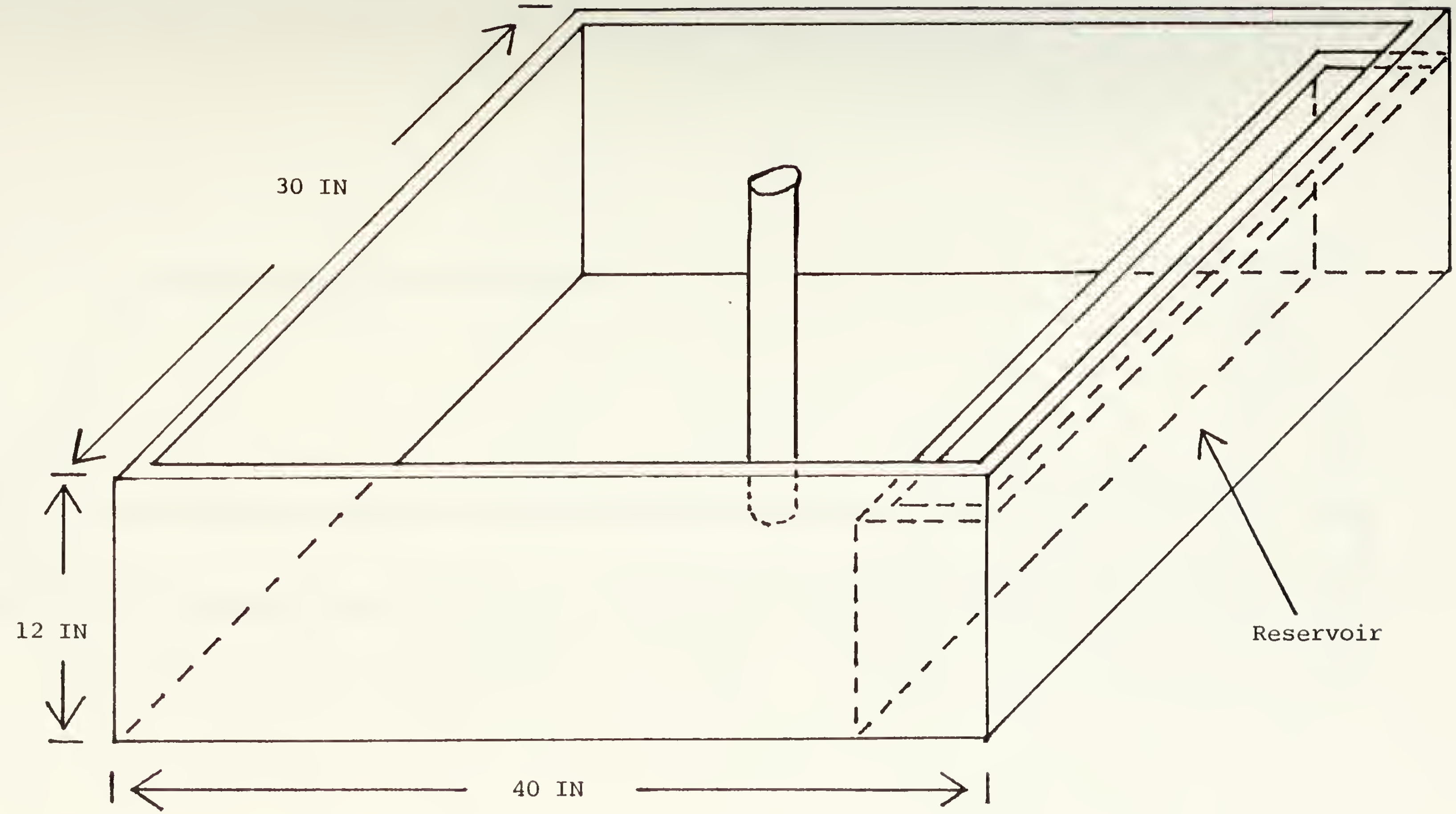

Figure 6. Container Design.

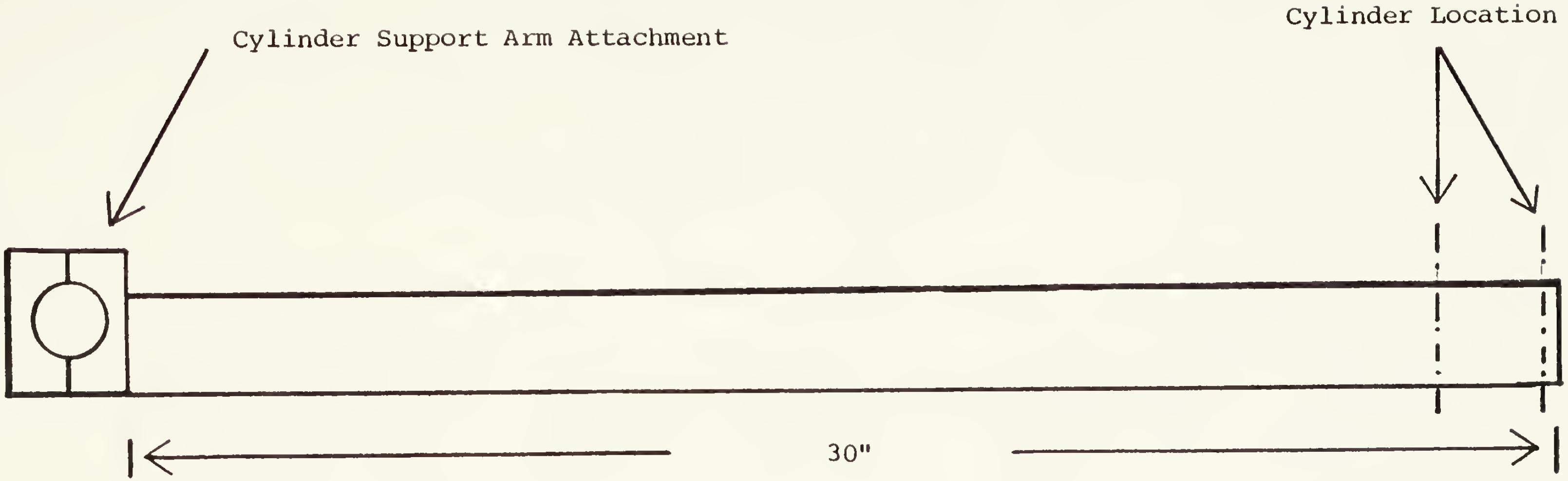

Figure 7. Cylinder Support Arm.

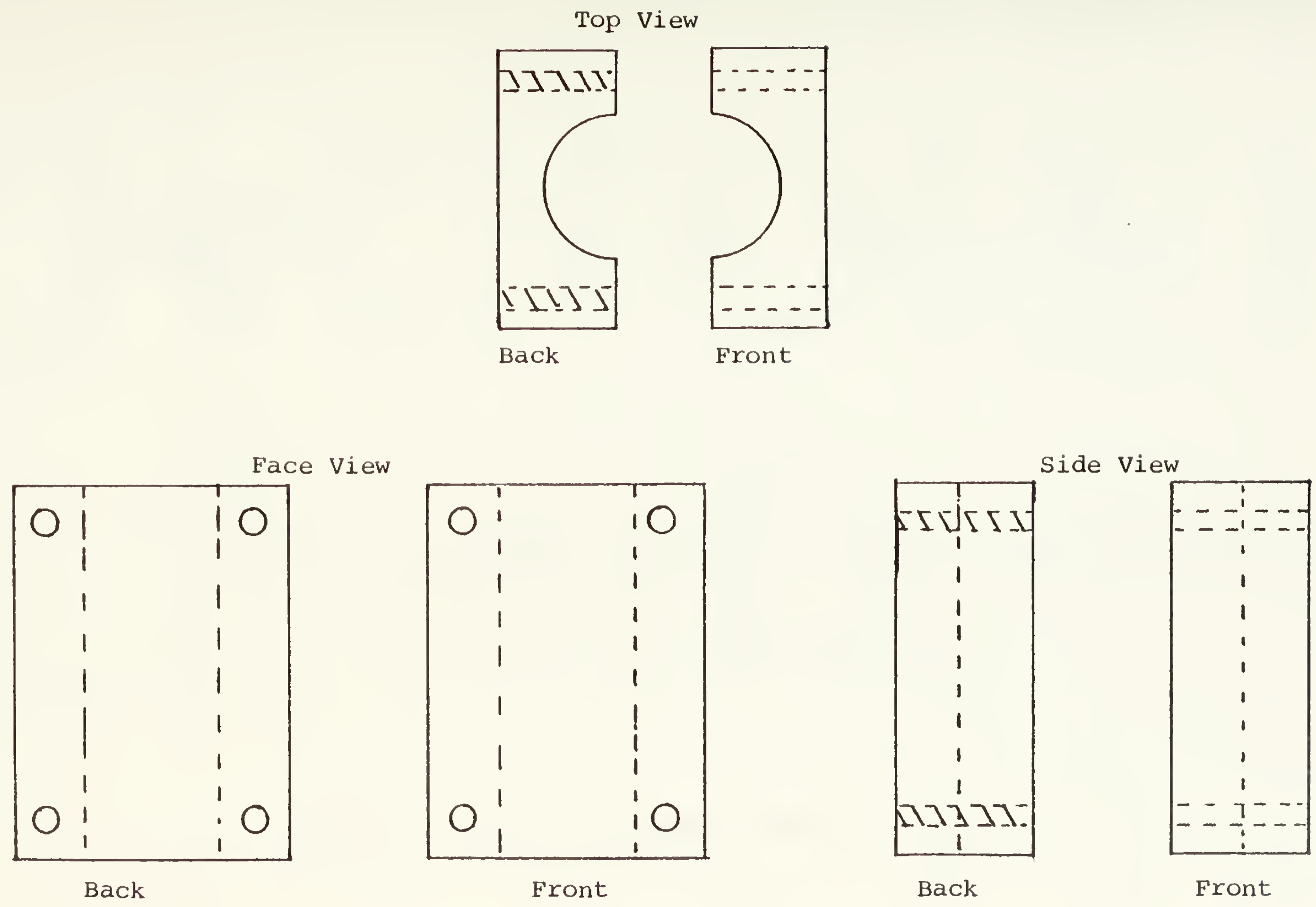

Figure 8. Cylinder Support Arm Attachments.

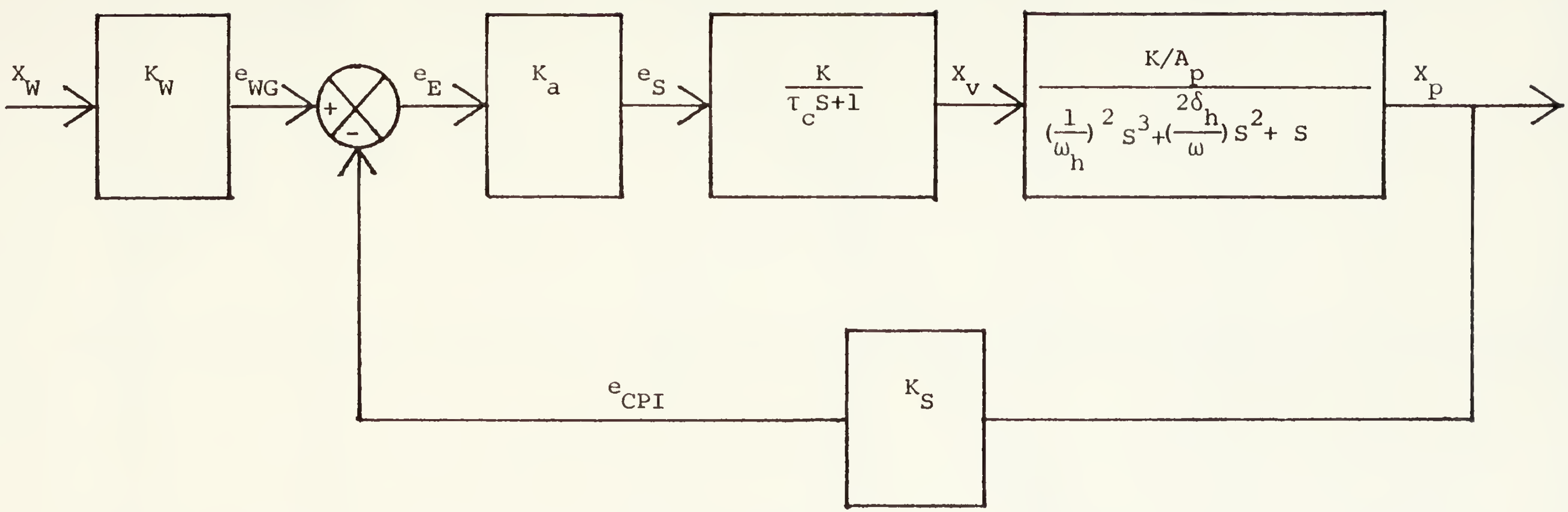

Figure 9. Closed Loop Block Diagram.

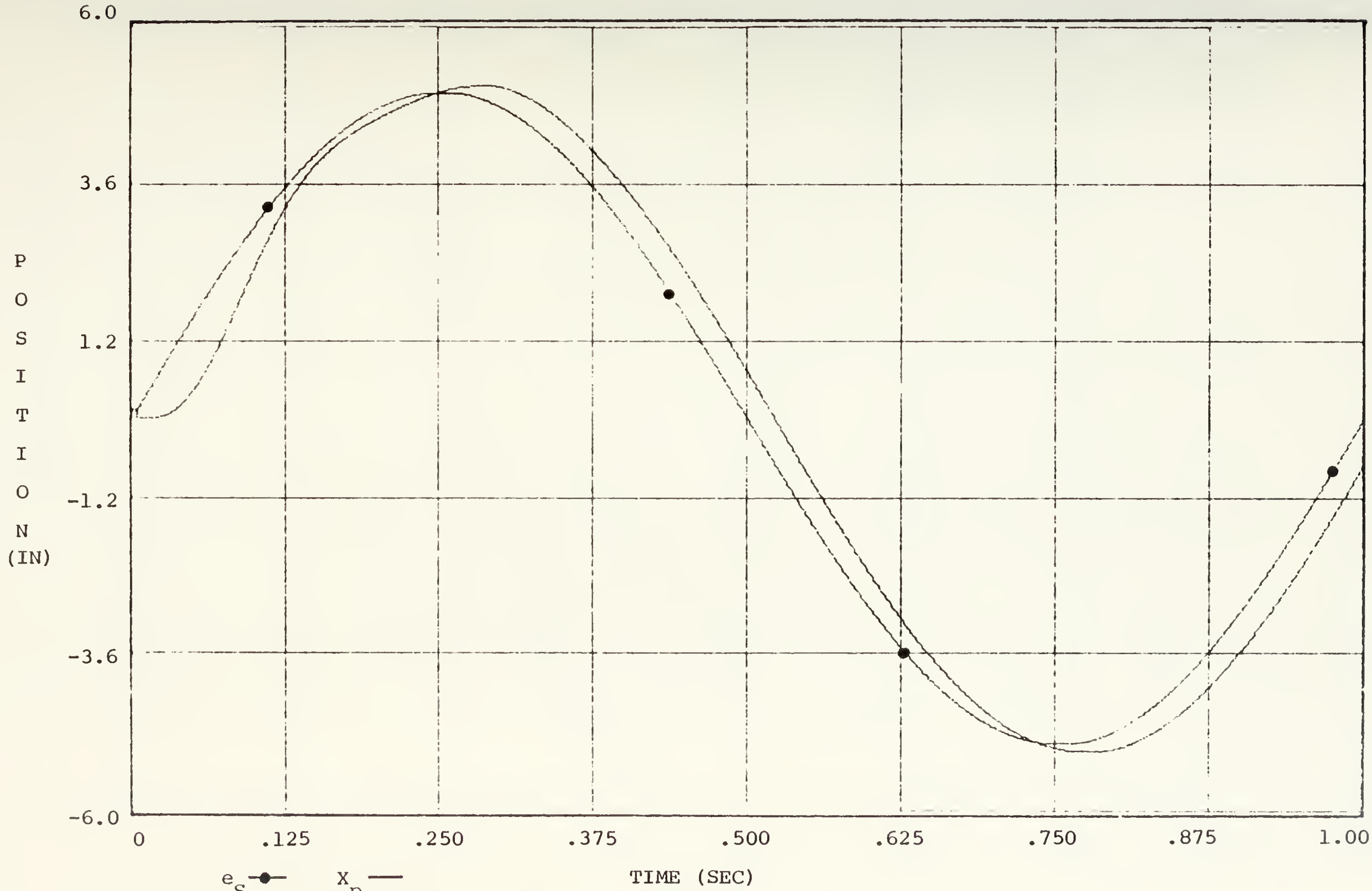

Figure 10. Phase Response at 1.0 HZ.

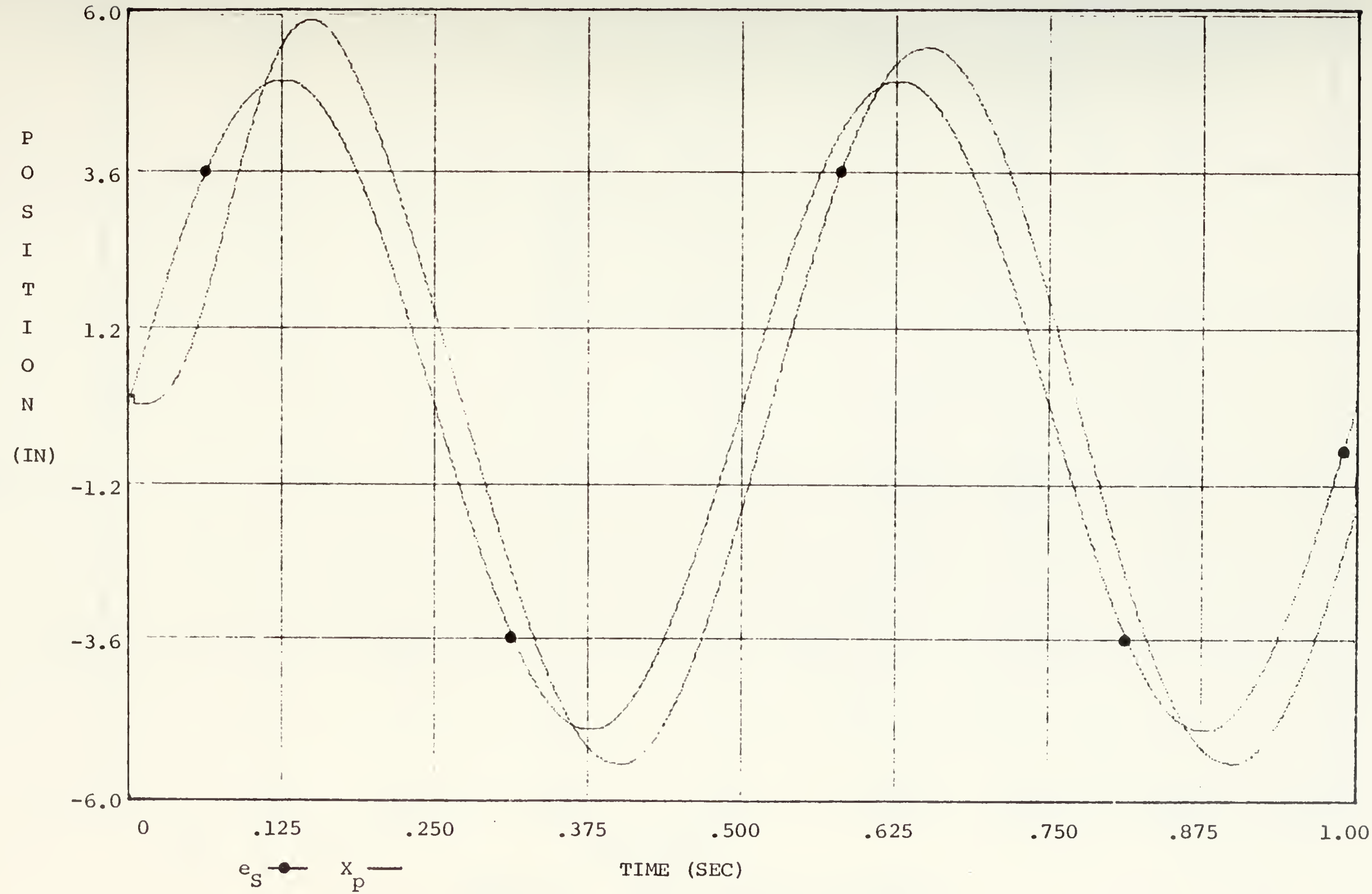

Figure 11. Phase Response at 2.0 HZ.

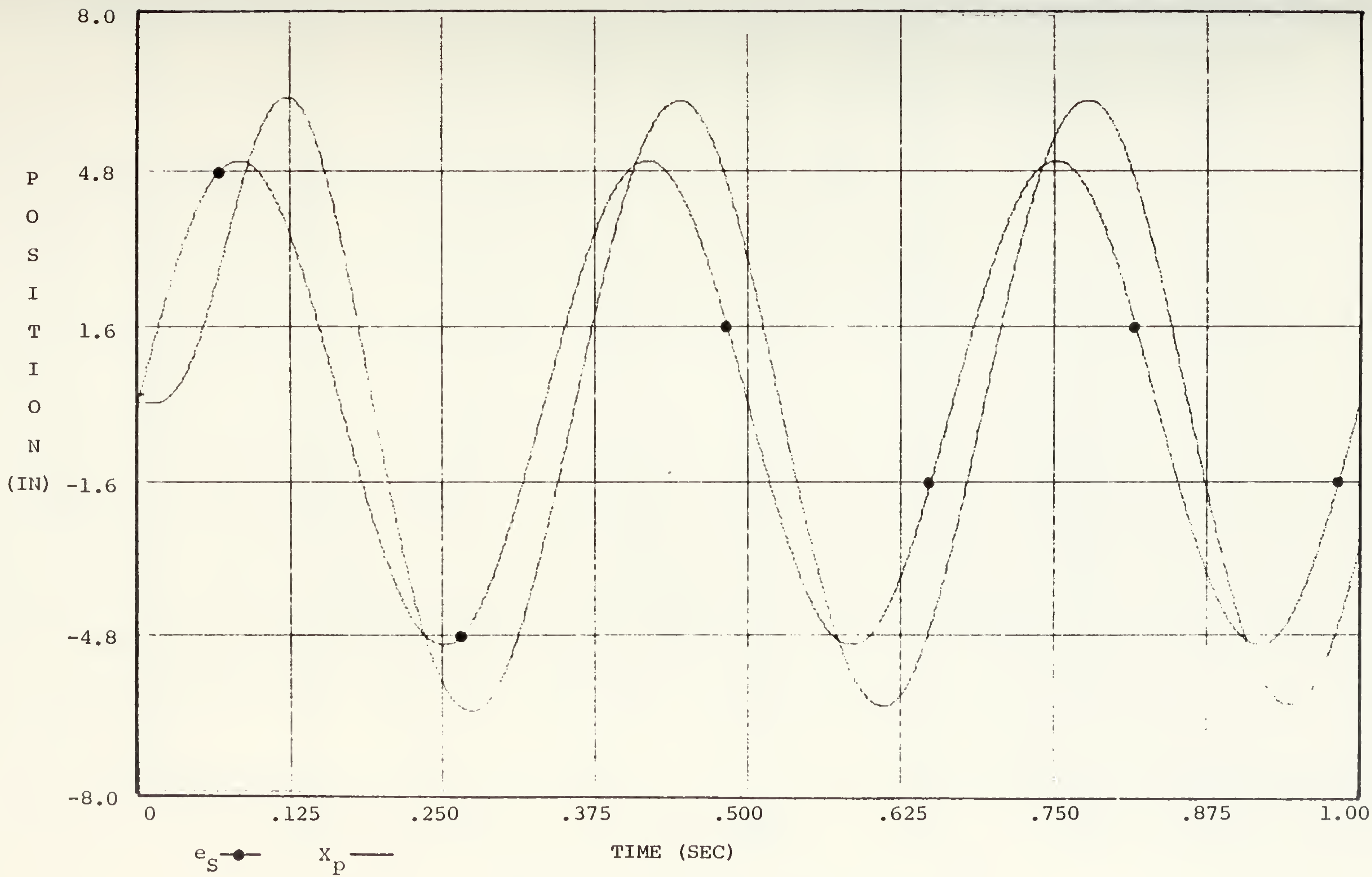

Figure 12. Phase Response at 3.0 HZ.

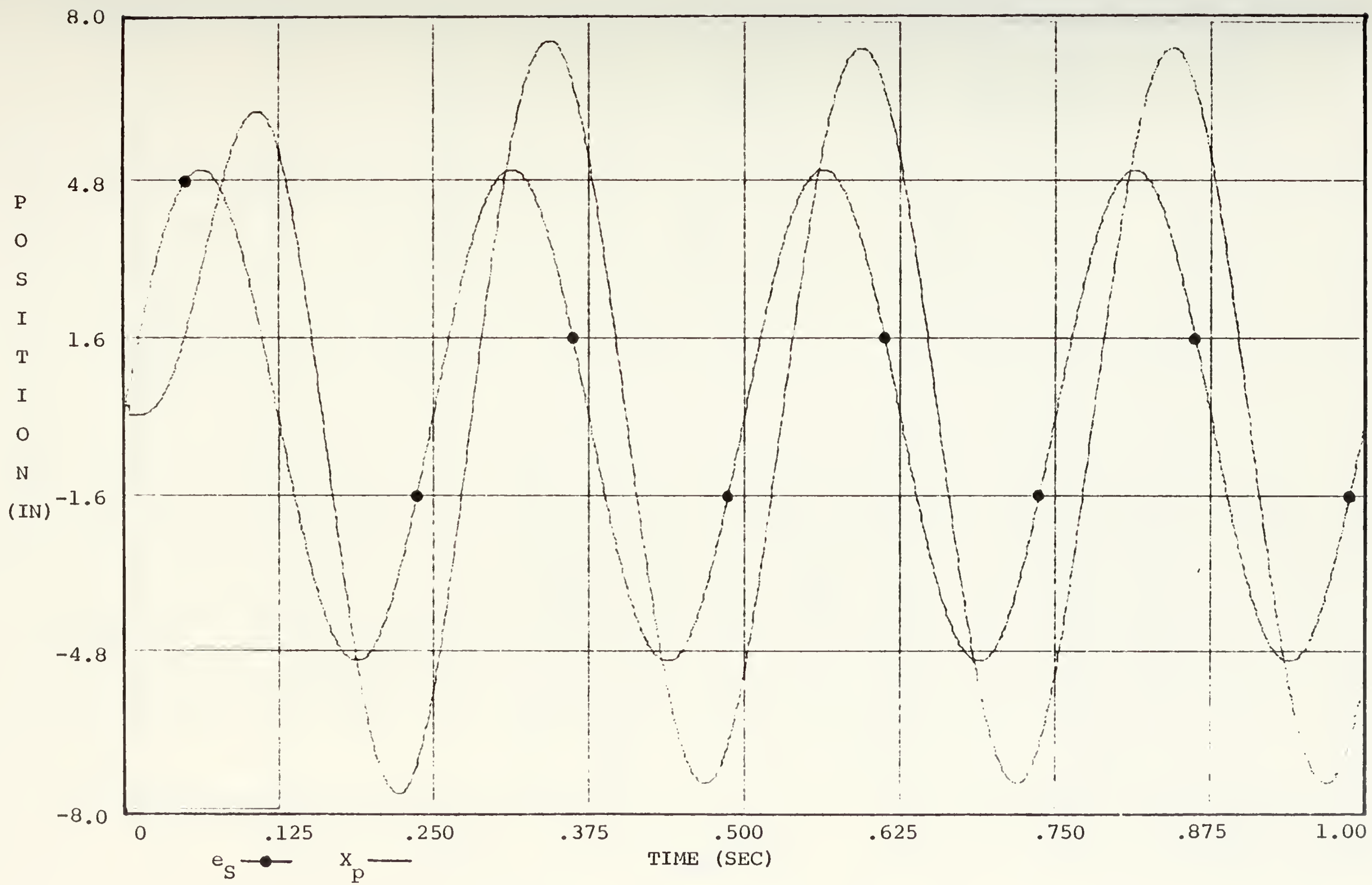

Figure 13. Phase Response at 4.0 HZ.

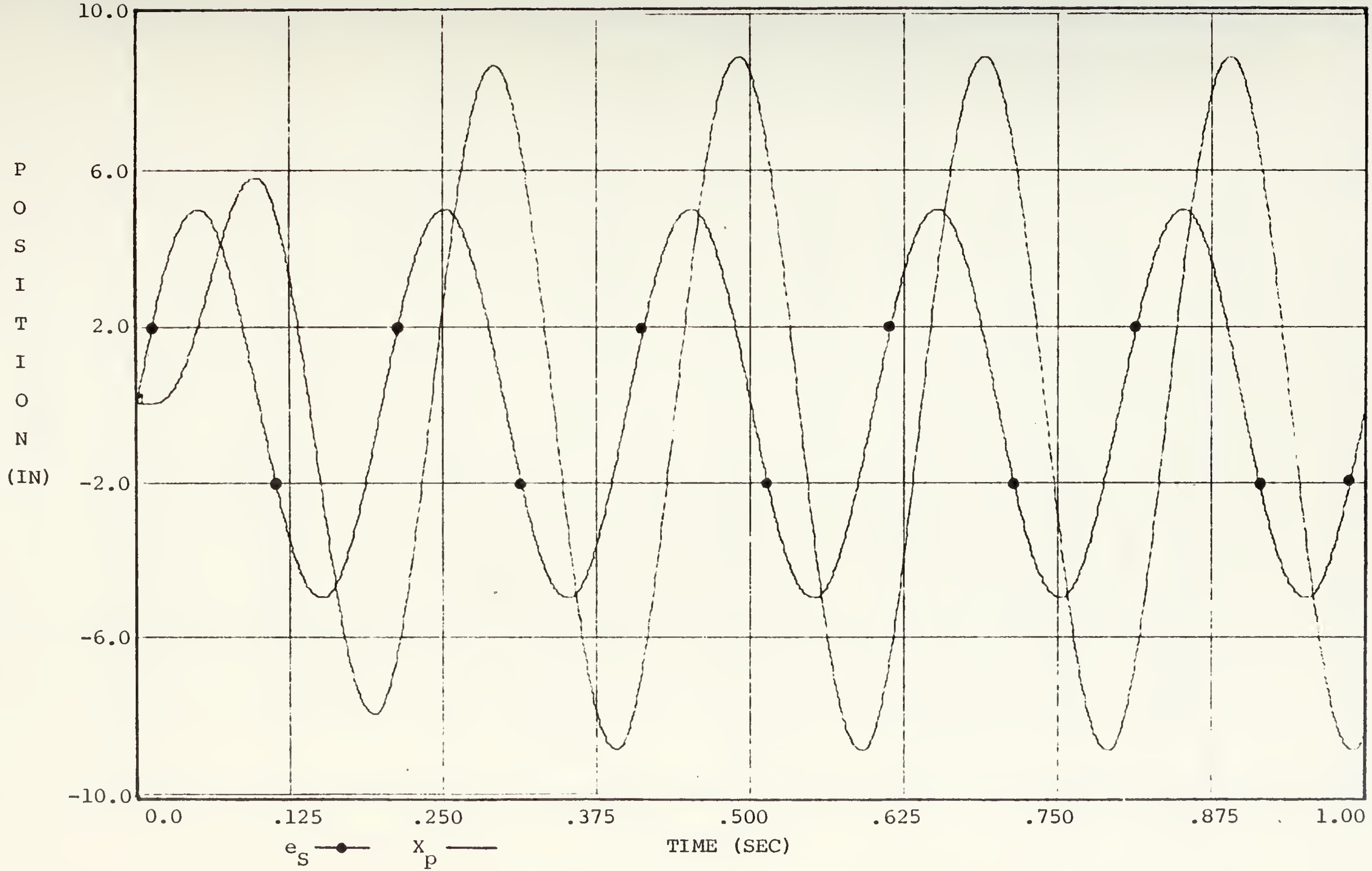

Figure 14. Phase Response at 5.0 HZ.

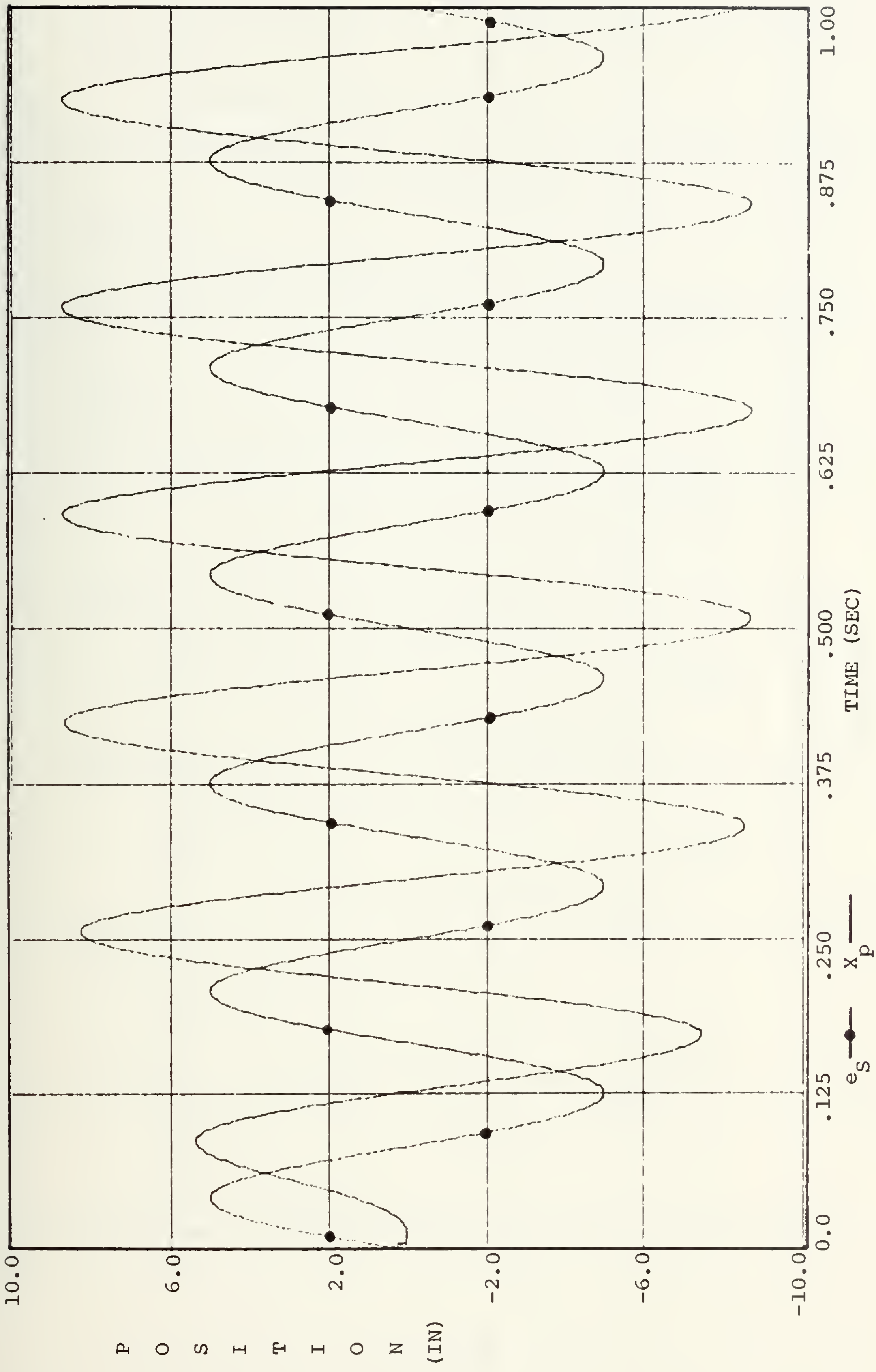

Figure 15. Phase Response at 6.0 HZ.

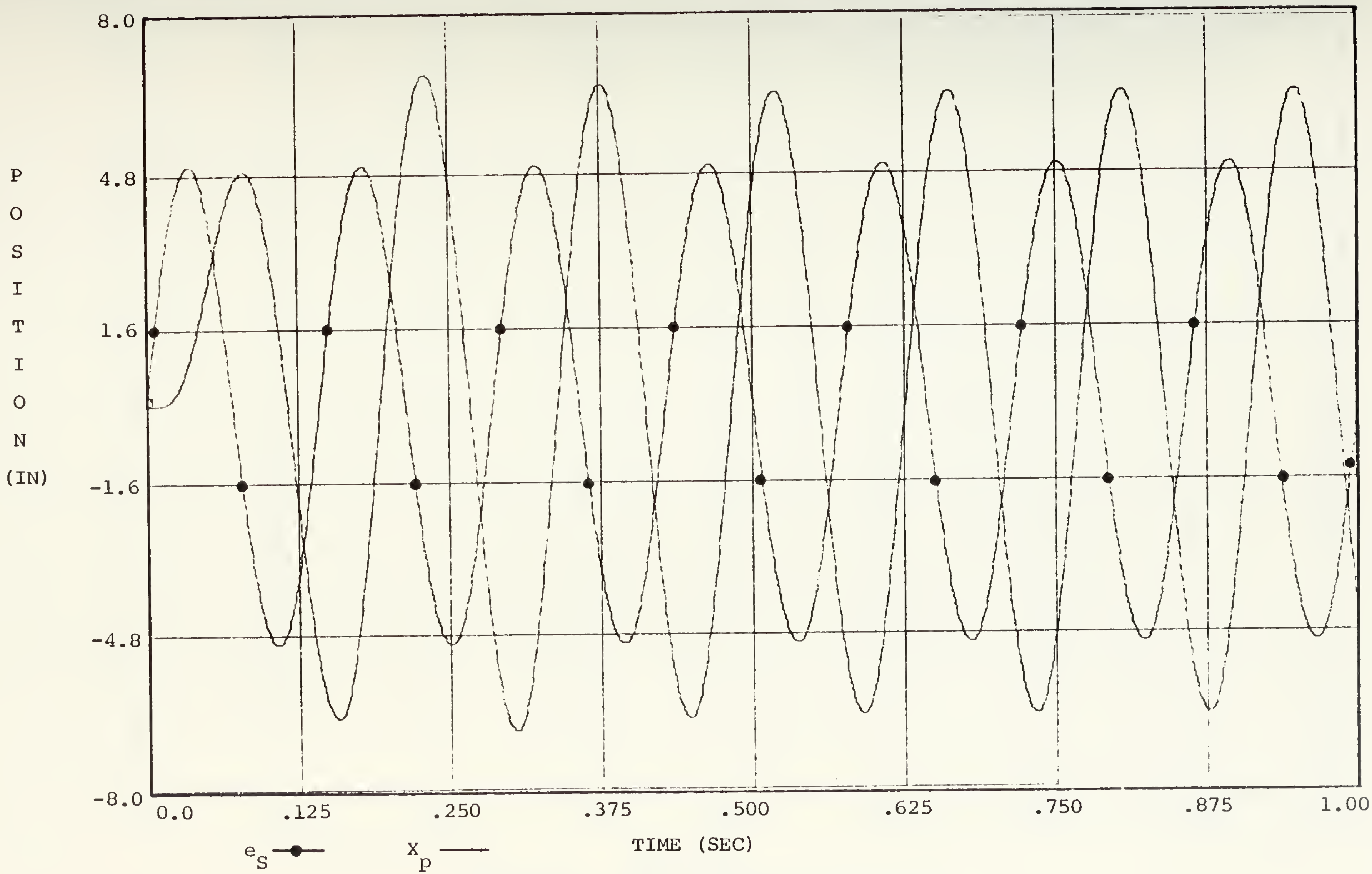

Figure 16. Phase Response at 7.0 HZ.

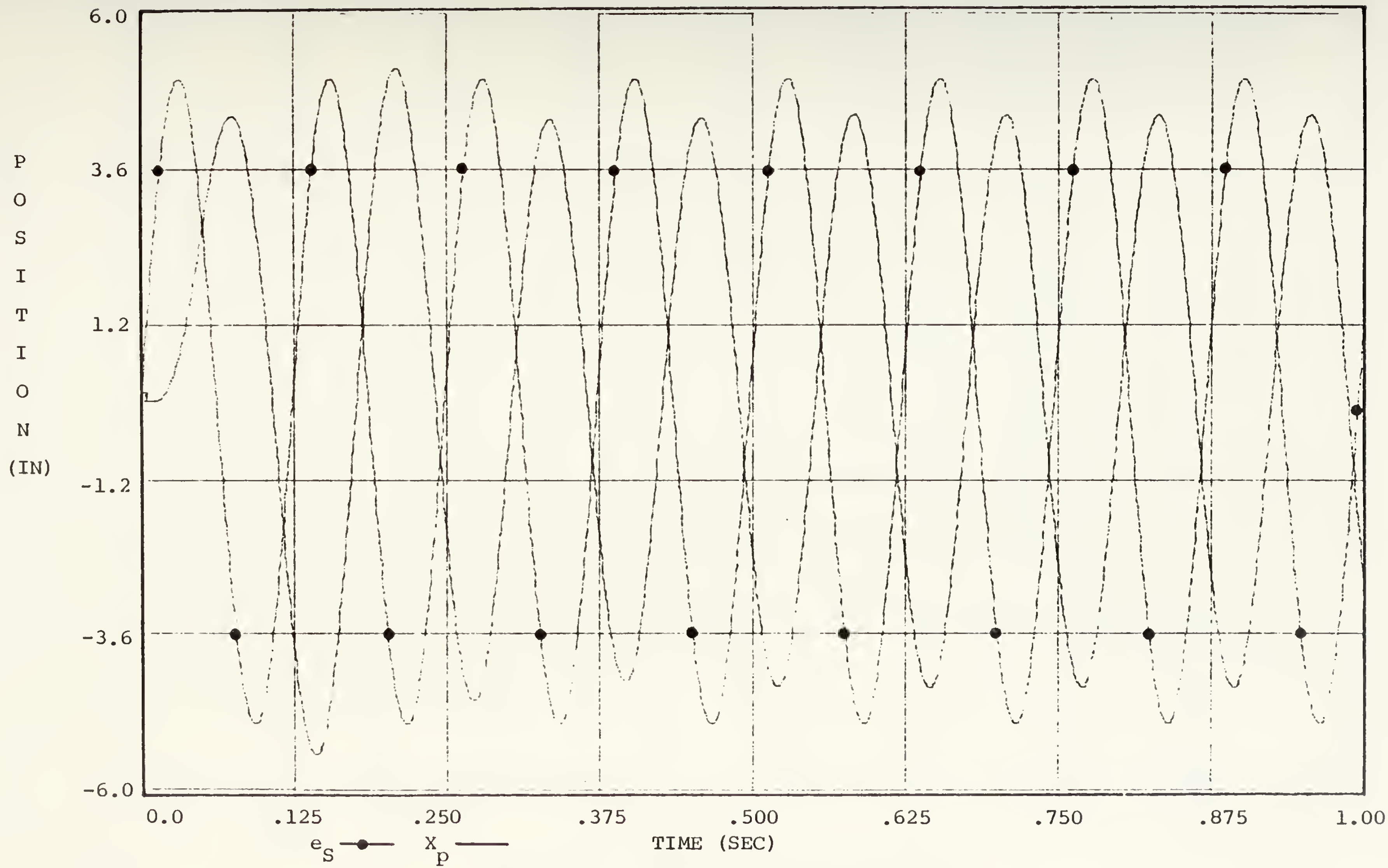

Figure 17. Phase Response at 8.0 HZ.

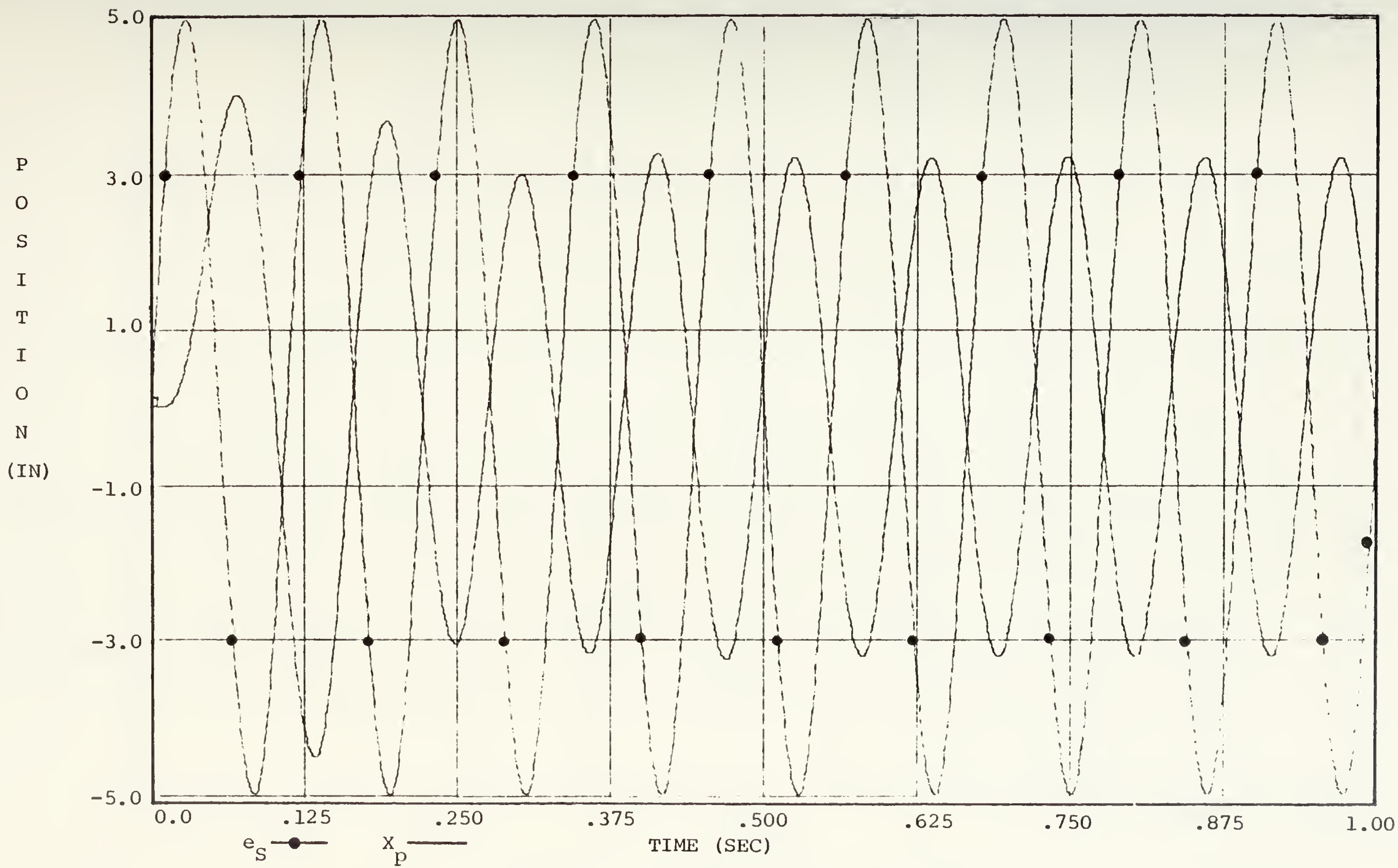

Figure 18. Phase Response at 9.0 HZ.

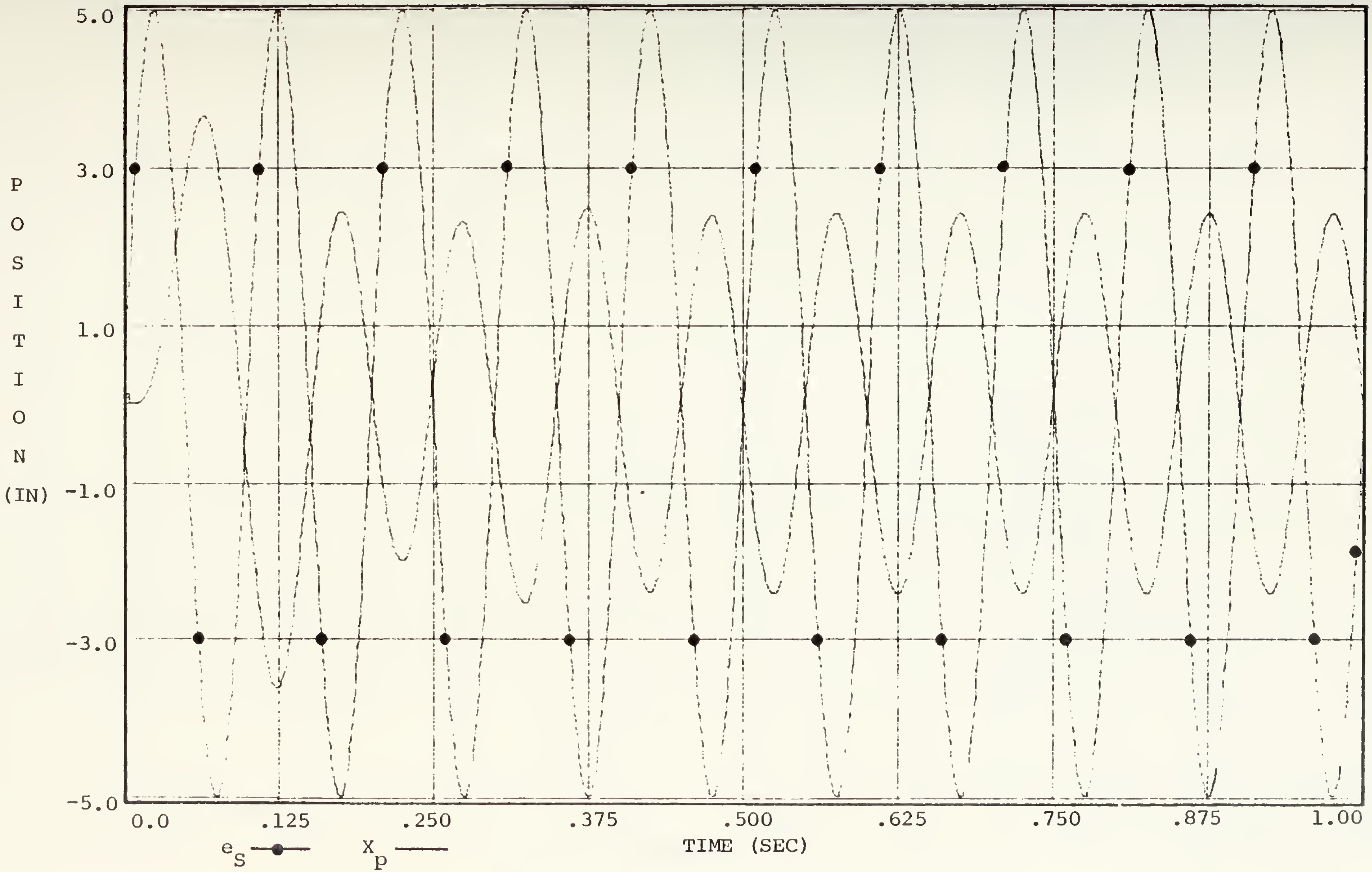

Figure 19. Phase Response at 10.0 HZ.

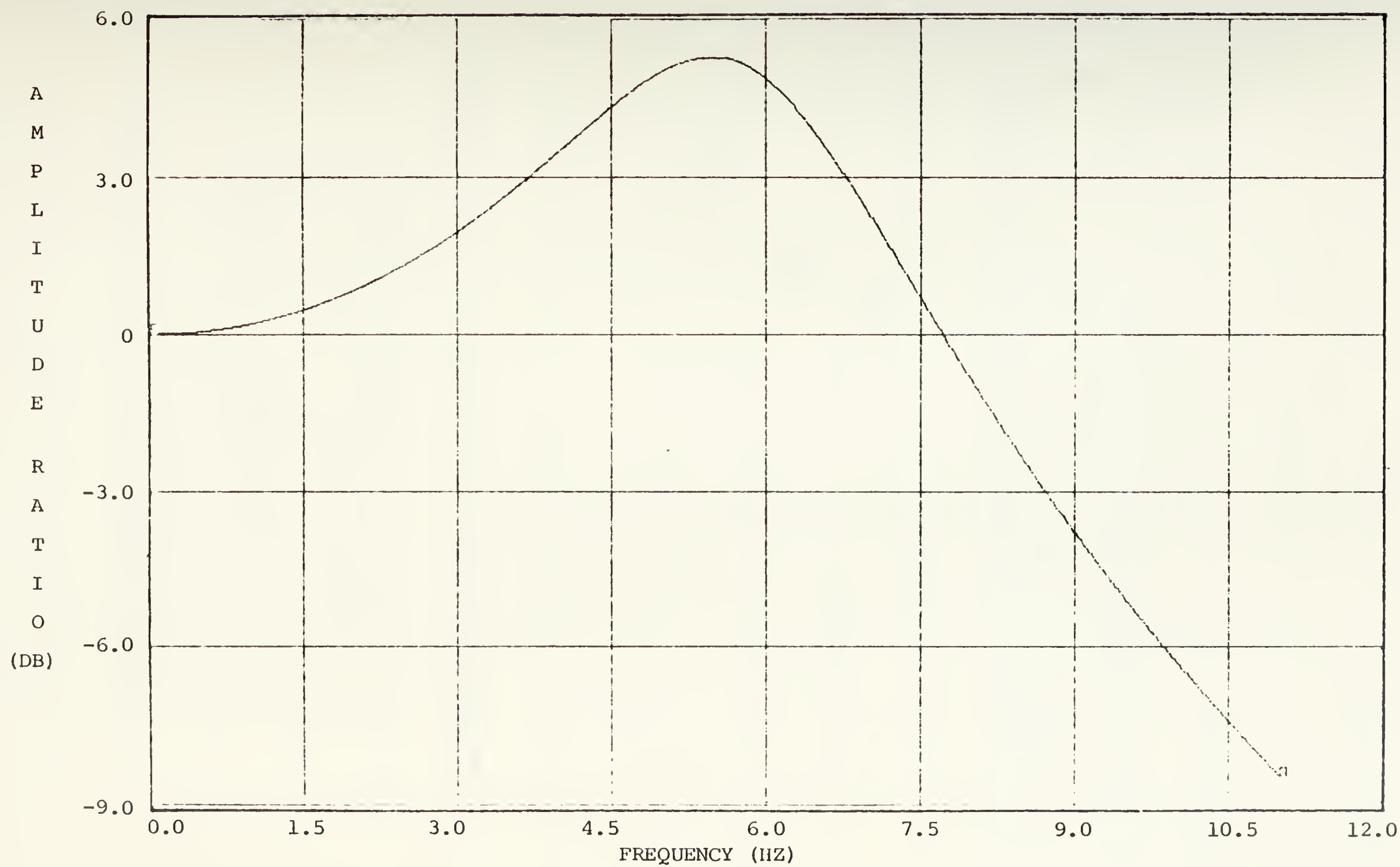

Figure 20. System BODE Diagram

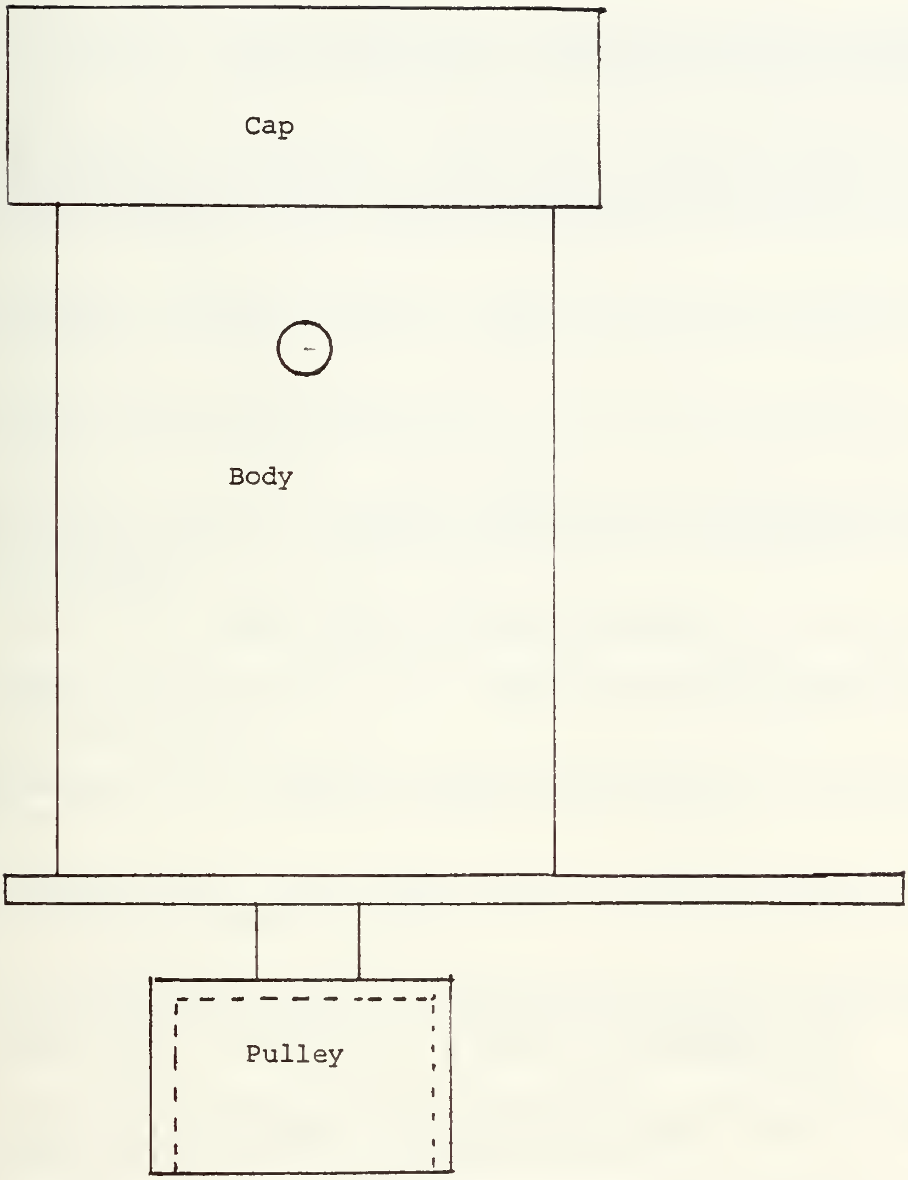

Figure 21. Cylinder Position Indicator Housing.

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